MACHINERY.

July, 1905.

VARIABLE SPEED MECHANISMS.—3.

CONE AND DISK DEVICES. (Continued.)

A device for transmitting motion was patented Dec. 6, 1887, No. 374,296, by G. Frank Evans, which was an improvement on the Laird patent No. 299,231. (Fig. 20, June issue, page 510.) The object of Mr. Evans' invention was "to provide a variable speed mechanism which might be applied to wood-working machines, or metal-working machines, or any machine that receives power through belting." The principal

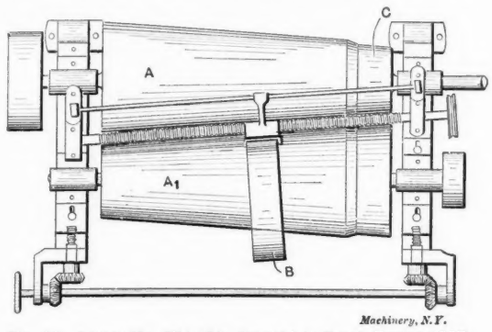


Fig. 22. Patent No. 374,296, Granted G. Frank Evans, 1887.

feature of novelty of the Evans invention appears to be the annular recesses at the right end of the cones A A_1 (Fig. 22) into which the endless belt B can be shifted so that motion may be stopped. In other words, the Evans invention provides means by which motion can be started and stopped by the same means which permits the speed ratio to be varied. He also provided a convenient means for varying the distance between the two cones; this device consisted broadly of two screws for moving the boxes of one cone, which were arranged so as to move in unison, being connected by a longitudinal shaft and bevel gears.

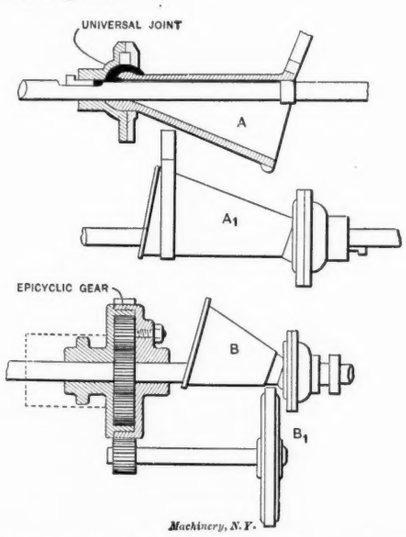


Fig. 23. No. 377,598, Issued to W. W. Beaumont, 1888.

Fig. 23 is part of one of the drawings illustrating the scope of patent No. 377,598, issued to W. W. Beaumont, February 7, 1888. The Beaumont device consists essentially of a hollow cone mounted on a shaft, with a universal joint at the small end so that the working side of the cone, or the side in contact with the belt, may lie parallel to the shaft axis. Since the cone moves at the same angular velocity as the shaft, it follows that all parts of the cone surface must move at the same angular velocity; and this means, of course, that the belt surface at the large end travels at a higher peripheral rate of speed than at the small end, although it apparently lies at the same radial distance from the shaft axis. When two of the Beaumont cones are to be used for variable speed transmission by means of belts, they are, of course, placed with the large end of one opposite the small end of the other as shown at A and A_1. The advantage claimed for this combination is, that the belt runs upon them very much as it does upon two cylindrical drums or pulleys. This contention, however, is not well grounded, for the climbing action of the

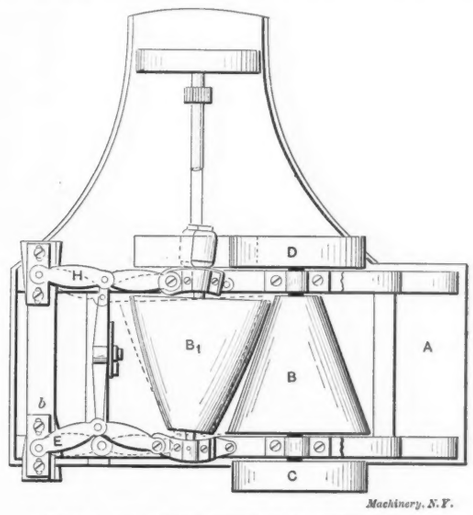


Fig. 24. No. 10,968, Granted to M. Hoffman, 1888.

belt upon the cones will be the same as though the cones were solidly mounted upon parallel shafts. The reason for this is, that the belt, as it first winds upon the cone, comes into contact with a surface that is nearer the base of the cone than that which it should properly occupy when it reaches the plane of the two shafts. The result is that there must be a continual tendency for the belt to shift to the large ends of the cones. The lower view in Fig. 23 shows one cone mounted on

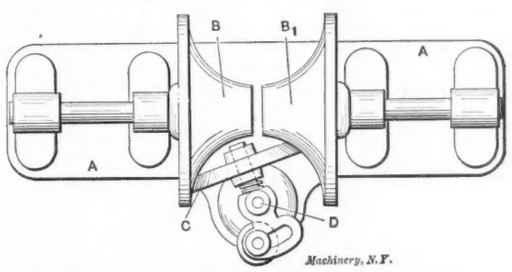


Fig. 25. Patent No. 457,100, Granted to C. C. Yates, 1891.

a shaft with an epicycloidal gear train, the outer gear of which is engaged with the pinion on the friction wheel shaft. The cone is movable longitudinally on its shaft so as to permit different cone diameters to be placed in contact with the friction wheel, and thus vary the rate of transmission.

Fig. 24 (patent No. 10,968, granted to M. Hoffman, November 27, 1888) embodies practically the same principle as that illustrated in Fig. 14 (June issue) with the difference that in the Reiss patent the rotating members are disk-shaped, while, in the Hoffman invention, they are more distinctly cone-shaped. One cone, B, is made with straight sides and cone B_1 is made with the sides curved so that only a short portion of its length can be in contact with cone B. This device which was intended as a feed-changing mechanism for the carriages

of saw mills, is provided with two toggle-joints, H and E, by which the boxes carrying the shaft of cone B_1 are shifted so that different portions of the two cones are brought into contact, thus varying the feed rate. This rather ingenious combination of toggle-joints enables a heavy pressure to be easily brought to bear on that portion of the cone which is in contact with cone B, and at the same time gives the swinging action necessary to bring different portions of cone B_1 into contact with cone B.

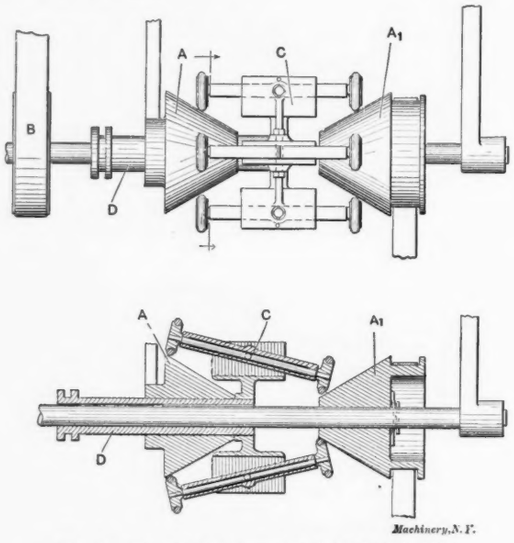


Fig. 26. No. 464,895, Issued to John Riddell, 1891.

Patent No. 457,100 was granted to Calvin C. Yates, August 4, 1891, for an improvement in frictional gearing. The device, as shown in Fig. 25, consists essentially of two shafts carrying hubs $B\,B_1$ on opposing ends, which are so shaped that a longitudinal section shows arcs of a circle, and the relative position of the two hubs is such that one arc is the continuation of the other, thus forming a partial torus in which the frictional member C may be shifted on the pivot D and still remain in frictional engagement with both B and B_1. The principle is very similar to the Hunt patent. (Fig. 18, June issue.)

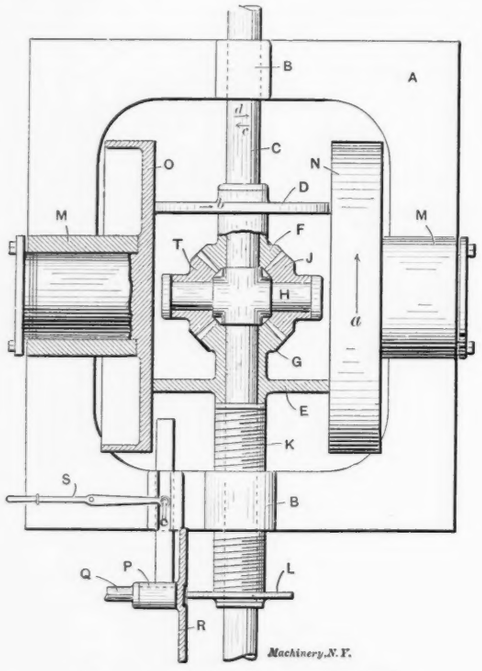


Fig. 27. No. 467,593, F. W. Gordon, 1892.

John Riddell was granted patent No. 464,895, December 8, 1891, for the variable transmitting mechanism shown in Fig. 26. This device consists essentially of two opposing cones, A and A_1, having coincident axes, one cone being, of course, the driving and the other the driven member. Motion is transmitted from one cone to the other by a series of friction rollers C, arranged in a circle. These rollers are connected by shafts and are provided with pivotal bearings in the center so that they are free to swing to whatever angular position is required by the position of the sleeve D. Moving D longitudinally changes the relative positions of the friction rollers upon the cones, and this, of course, changes the relative rate of transmission.

Patent No. 467,593, granted to F. W. Gordon, January 26, 1892, is described in an interesting specification and is illustrated in Fig. 27. D and E, referred to as brush wheels, are mounted loosely on shaft C and have projecting hubs which terminate in the bevel gears, F and G. Between these gears are other bevel gears, T and J, mounted on studs projected at right angles from the shaft, thus forming the familiar "jack-in-the-box." Brush wheels D and E are held in frictional contact with the disks N and O. The mechanism shown below is for varying the position of the brush wheels upon the disks. Quoting from the specification, "Assume that disk N is driven, as by belt, at high speed in the direction of the arrow, a, at a regular rate of speed. Assume that the brush-wheels, D and E, are equidistant from the center of the disk, as they appear in the drawing. The disk will drive brush-wheel D in the direction of the arrow, b, and will drive brush-wheel E in the opposite direction, both brush-wheels revolving at the same speed. Gear F turns with brush-wheel D and rotates the planetary gear, J, upon its stationary stud, H. Gear G forms no impediment to the rotation of the planetary gear, because gear G is turning in the same direction and at the same rate of speed as it would if driven by the rotation of the planetary gear instead of through the brush-wheel E. The result is that the shaft stands still and the planetary gear turns idly on its stud. If now we move the shaft endwise a trifle toward the upper portion of the drawings, we still have the two brush-wheels D and E turning in opposite directions, as before; but the two brush-wheels no longer engage the disk equidistantly from its center, and therefore the two brush-wheels will not turn at their former rates of speed, nor will they both turn at the same rate of speed. Brush-wheel D will turn faster than it did before and brush-wheel E will turn slower than it did before. Gear F still seeks to turn the planetary gear

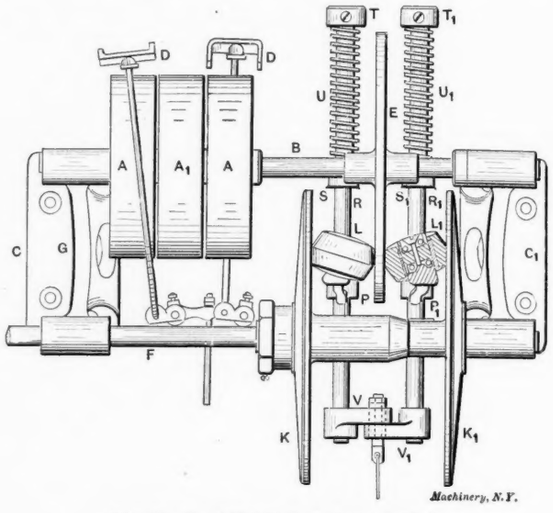


Fig. 28. No. 568,367, E. C. Osgood, 1896.

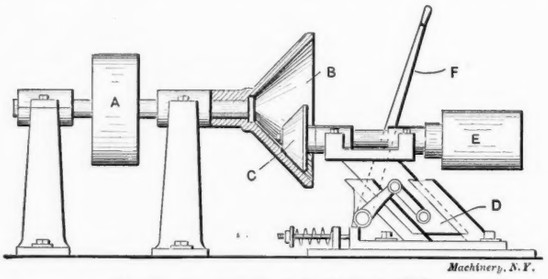


Fig. 29. No. 599,193, E. R. Plaisted, 1898.

idly on its stud; but gear *G* is not turning fast enough to allow the planetary gear to rotate freely, as in the former case. If the shaft could not turn, a deadlock would occur at the planetary gear; but the shaft is free to turn in the direction in which brush-wheel *D* turns (arrow *b*) and permit the planet-stud and planet-gear to take up a motion of planetary advance around the shaft. The rate of this planetary advance will be

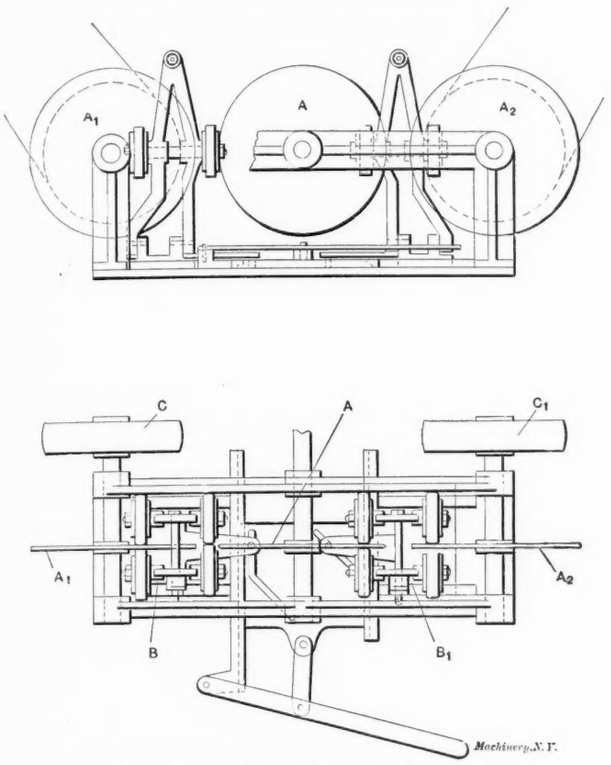


Fig. 30. No. 650,815, Granted to James Arthur, 1900.

governed by the difference in velocities between gears *F* and *G*. In other words, if the two brush-wheels *D* and *E* do not turn at the same velocity, the planet gear is forced into planetary motion, and consequently the shaft *C* is rotated in the direction of rotation of the fast-moving brush-wheel. The two brush-wheels may be so adjusted that their rates of rotation differ to an infinitely small degree, and the result will

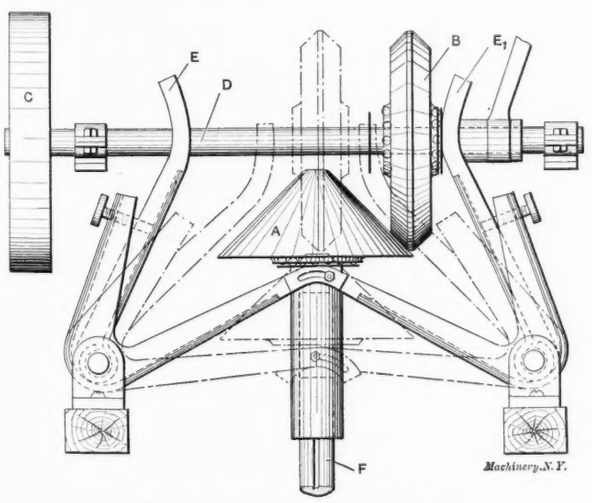


Fig. 31. No. 637,202, J. C. Des Granges, 1899.

be that the shaft is turned at an infinitely slow rate of speed. The rate of speed of the shaft may be altered by adjusting the position of the brush-wheels with reference to the disk which drives them, and the direction of motion of the shaft is also under control, the shaft moving in the direction of rotation of the faster-turning brush-wheel. The device therefore provides for an infinitely great reduction of speed, for a control of the direction of transmission, for the most delicate graduations of reduction, *and for powerful transmissions.*

It is scarcely necessary to point out the mistake of the inventor in assuming that this device can be employed for a powerful transmission. In other words, the change in velocity ratio is not accompanied by the corresponding change in the leverage. The actual torque of shaft *C* must always depend upon the friction coefficient of brush-wheels *D* and *E;* hence, it is obvious that the device is not suitable for "powerful transmissions."

In the mechanism for transmitting motion by variable speed, patented by E. C. Osgood, September 29, 1896, No. 568,367, (Fig. 28), the inventor claims a great range of speeds and exceed-

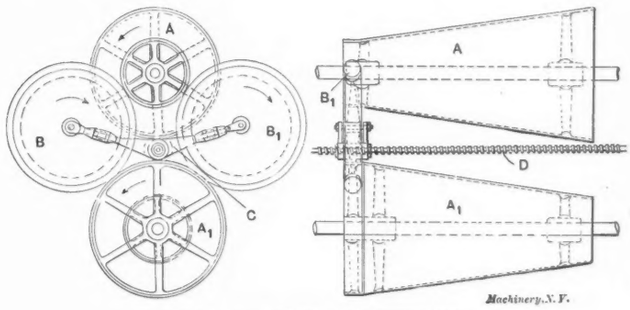


Fig. 32. No. 675,133, A. Janssens 1901.

ingly exact adjustment, with slight friction and with great economy. It comprises a shaft *B* on which is mounted the disk wheel *E*; and the two disk wheels *K* and K_1, arranged on opposite sides of *E* and embracing between them the rollers *L* and L_1. These rollers are mounted on ball-bearings and are of double cone shape to minimize as far as possible the difference in speed of the actual surfaces in contact, from which it follows that the friction rollers are tipped so that the planes of rotation are at angles other than right angles with the plane of rotation of the disk wheels.

In the invention shown in Fig. 29, which was granted to E. R. Plaisted, February 15, 1898, No. 599,193, the hollow cone *B* is employed to drive friction cone roller *C*. The cut shows the means employed for varying the relative position of *C* and *B*, and thus varying the velocities of pulleys *A* and *E*. One interesting feature of this invention is, that when *C* is brought to the center of cone *B*, the two run at the same rate, the action in this position being essentially the same as that of a cone clutch.

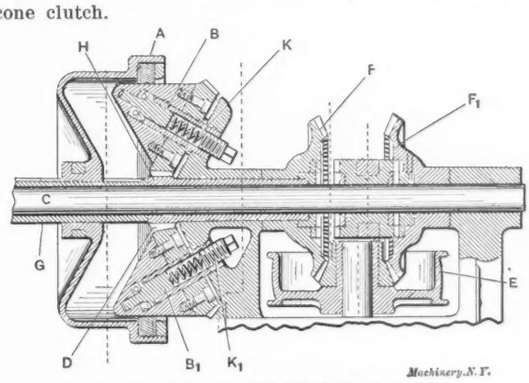


Fig. 33. Patent No. 675,669, O. Norling, 1901.

Fig. 30 illustrates the invention of James Arthur, patented June 5, 1900, No. 650,815, which was designed for transmitting considerable power. It also provides means by which the driven disk *A* may be reversed at will. When made with the reversible feature, the power is transmitted through pulleys *C* and C_1 which drive disks A_1 and A_2 respectively. The nests of friction rollers, *B* and B_1 are mounted on slides, and connecting these slides is a slotted plate which is given a transverse motion by a hand lever. The slots in this plate are so shaped that one slide remains stationary during the movement of the other slide in or out with its rollers in engagement with its respective disks. Suppose that the slide is being operated so that the nest of friction rollers B_1 is being moved to the right, the left-hand slide remains station-

ary until friction rollers B_1 have moved out of engagement with disk A and then the movement of the slide carrying the friction rollers B begins and effects the reversal of A.

Of a far less practical nature is the invention, Fig. 31, patented by J. C. Des Granges, November 14, 1899, No. 637,202, as a motion-reversing and speed-changing mechanism. Means are provided for shifting cone A longitudinally upon the shaft F, and simultaneously shifting the double cone B on shaft D so as to maintain frictional contact between the conical sur-

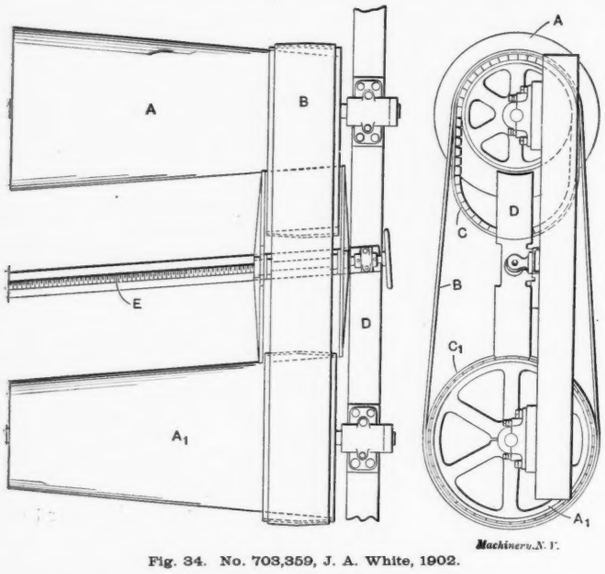


Fig. 34. No. 703,359, J. A. White, 1902.

faces. By continuing the motion of B beyond the center, it is evident that reversal of motion would be obtained and that the curved arms would in theory sustain B in contact with A in the reversed position.

The variable speed transmitting gear shown in Fig. 32 was patented May 28, 1901, No. 675,133, by A. Janssens. It comprises two cones A A_1, mounted on parallel shafts in such a manner that the apex so called, of one cone corresponds with the base of the other, and connected by the friction rollers B

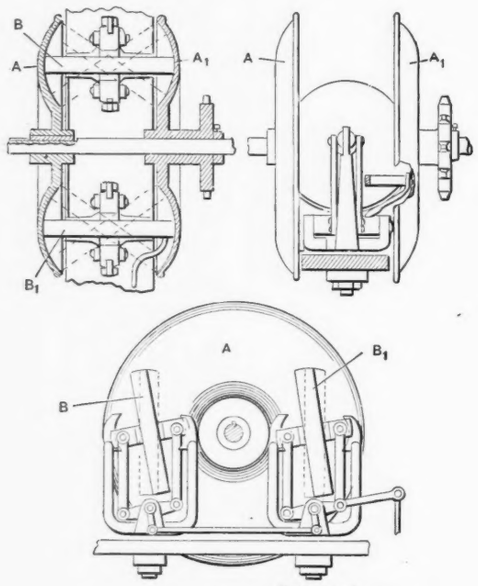


Fig. 35. No. 703,237, E. P. Cowles, 1902.

and B_1. These friction rollers are connected by a jointed frame C and are given a longitudinal motion by means of screw D. The inventor states that his object is "to remedy certain defects of the transmitting gear heretofore encountered, by permitting the use of rollers of convenient diameter, corresponding to the power to be transmitted by the mechanism. At the same time, the invention enables the constructor to double the number of points of contact without difficulty and to balance as completely as possible the strain acting on the principal parts of the transmitting device."

The invention illustrated in Fig. 33 shows one drawing illustrating the specification of patent No. 675,669, issued to O. Norling, June 4, 1901. This speed change gear was designed more particularly for use in driving the feed mechanism of automatic metal working machines; hence the peculiar disposition of the parts as shown in the cut. The clutch mechanism shown between the bevel gears F and F_1 is evidently for the purpose of reversing shaft C, and it does not affect the function of the variable speed device. The hollow friction wheel, A, which is mounted on a sleeve G, is splined thereon so that it may be moved longitudinally to vary the speed motion. Within a hollow wheel are mounted the adjustable friction cone rollers B and B_1, which have bevel gears fixed to their bases. These gears mesh with the central gear H, which is part of a sleeve, on which is mounted the bevel gear F; hence it is plain that the rate of motion transmitted to sleeve G will depend upon the relative position of A upon it, since motion is communicated to G from the belt pulley E through bevel gear F, gear H, and friction rollers B and B_1. The pressure of the friction rollers on A is regulated by the adjusting screws and springs K K_1.

In the speed changing device shown in Fig. 34, which was patented by J. A. White, June 24, 1902, No. 703,359, the principal feature of interest is the provision made for the supporting of the driving belt upon the cones and compensating the effect of the taper. The inventor says that "Heretofore the

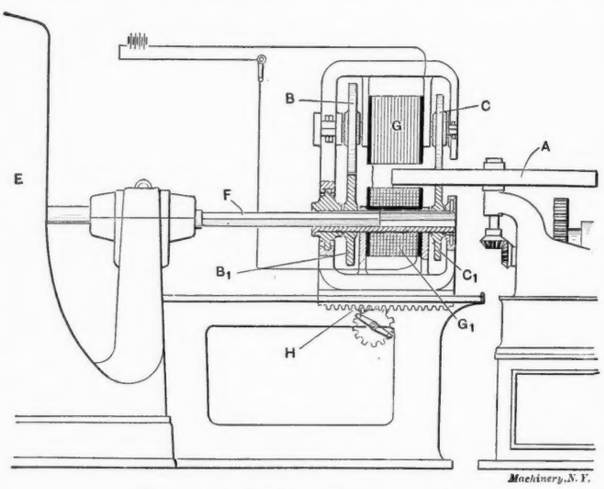


Fig. 36. No. 745,593, G. H. Gibson, 1903.

power which could be transmitted from a driving to a driven element through cone-pulleys was limited owing to the fact that a wide belt could not be used, the width of the belt being limited because of the taper of the pulley or pulleys." Hence he provides an inner band for each cone-pulley beveled on the inner face, according with the bevel of the cone pulleys, and possessing an inner face susceptible of longitudinal expansion and contraction, which is called a compensating band. The illustration shows the means employed for traversing the belt B from one end of the cone-pulleys to the other and also shows the compensating bands C C_1. In the case of C, which is at the small end of the cone, only a portion of its length is required to encircle the working periphery of the cone, but in the case of C_1, which is at the large end of the cone, the full length is required. The invention is, in effect, a combination cone-pulley and expanding pulley device.

Fig. 35 shows a variable speed friction gearing patented by E. P. Cowles, June 24, 1902, No. 703,237, which provides means for automatically shifting the position of the frictional rollers relative to the driving and driven elements in order to vary the speed. The inventor uses the word "automatically," but his meaning is that his invention provides means by which the position of the friction rollers may be shifted with a minimum exertion of the power on the part of the operator. He mounts the rollers B and B_1 between the disks A and A_1 so that they may be adjusted from their normal tangent posi-

tion to a position in which other planes of rotation will not be tangent to a circle concentric with the shaft. In this position they tend to shift at a rate according to the angle to which they are twisted by the operator, since their faces tend to travel in a spiral within the torus formed by the two disks A and A_1.

Patent No. 745,593 was granted December 1, 1903, to G. H. Gibson for a magneto frictional gearing. The driving shaft F from the motor E is splined and drives a sleeve, on which are mounted the rollers C_1 and B_1. Between these rollers is mounted a spool of wire G_1 and at B, G, and C is a similar arrangement. The coils G and G_1 form an electromagnet which, being energized, forces the rollers C and C_1 against the sides of the disk-wheel A; the energizing force also brings rollers B and B_1 into forcible contact, thus transmitting motion to C and making it a driver as well as C_1. The rack and pinion

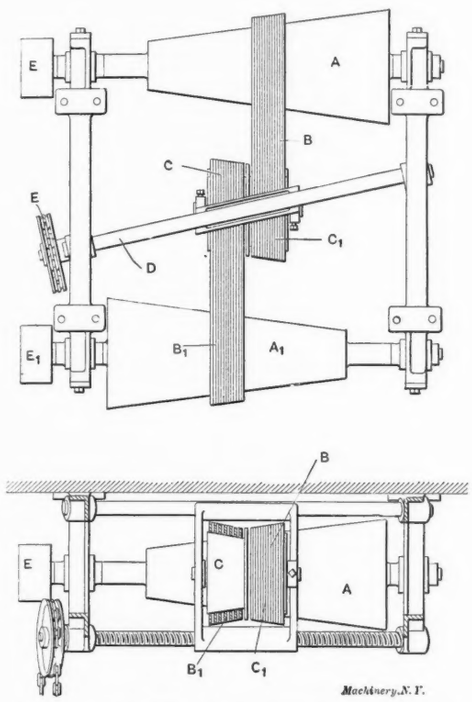


Fig. 37. No. 759,872, Wm. Evans and P. W. Knauf, 1904.

provided for shifting the magnetic friction rollers in and out upon A for varying the speed are shown at H.

The variable speed transmission device illustrated in Fig. 37 was patented May 17, 1904, No. 759,872, by William Evans and Paul W. Knauf. In this invention the cone-pulleys A and A_1 are not directly connected by one belt, but two belts are used which transmit motion from one to the other by means of a double cone-pulley $C\,C_1$. The object of this arrangement is to compensate for the taper of the cones so that all portions of the belts may travel at the proper speed and thus avoid the destructive action due to the cone and belt elements traveling at different rates of speed. The invention therefor is supposed to make it possible to use wide belts and thus transmit considerable power.

* * *

The *Journal* of the Worcester Polytechnic Institute abstracts an item from a French contemporary giving the efficiencies of various kinds of gears of the proportions used in automobile service, as follows: Spur gearing, steel on steel, greased, but exposed to street dust, 90 and 80 per cent; spur-gearing having steel pinion engaging with fiber gear-wheel, 88 and 80 per cent; spur-gearing having leather pinion engaging with a cast-iron gear-wheel, 88 and 80 per cent; spur-gearing, steel on steel, in oil bath, 92 and 90 per cent; roller chain, lubricated and exposed to the air, 94 and 92 per cent; and universal joint, 95 per cent. In the above cases the first set of figures for each gear represents the percentage of efficiency for new gearing, and the second set for worn gearing.

CALCULATIONS FOR THE SHAFT, GEAR AND BEARINGS OF CRANE MOTORS.

GFORGE J. LEIRE.

To illustrate definitely the use of the table and diagrams in the method of calculation about to be set forth the following example will be taken:

Given: A crane motor with 4 poles, 15 H. P., 750 R. P. M. at normal load:

Diameter of armature..... = 9½ inches
Air gap ..... = 3/32 inch
Area of pole face..... = 29 square inches
Density in air gap, given in lines of force per square inch..... = 55,000
Total weight of rotating part..... = 150 pounds

From a general layout drawing of the motor we have the dimensions given as in Fig. 1.

Motors for hoisting purposes are usually series wound, and thus run at different speeds under different loads. Therefore, if the motor is to be run with a certain over-

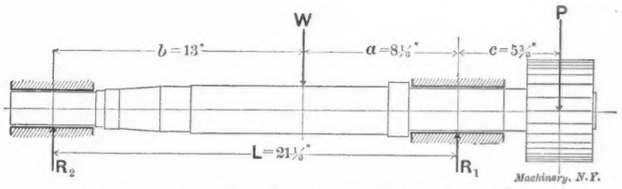


Fig. 1. Dimensions of Shaft and Bearings, from General Layout.

load, we have to take into consideration the corresponding speed and density in air gap, which can be obtained from the speed and excitation curves of the motor. In this example we suppose an overload of 25 per cent, and have accordingly:

18.75 H. P., 700 R. P. M. and a density in air gap = 58,800 lines of force per square inch.

Calculating the Gear.

For figuring the gear we suppose a diametral pitch = 5 and number of teeth = 18.

This gives us a pitch diameter $= \frac{18}{5} = 3.6$ inches.

Thus the pitch line speed at

$$700 \text{ R. P. M.} = \frac{\pi \times 3.6 \times 700}{12} = 660 \text{ feet per minute.}$$

It is not advisable to use a pitch-line speed exceeding 1,000 feet per minute, on account of noisy running.

Before figuring the width of gear we have to determine the pressure P on the teeth. This is given by the following formula:

$$P = \frac{\text{H. P.} \times 33{,}000}{\text{Pitch line speed}}$$

where P is expressed in pounds and pitch line speed in feet per minute. Thus for 18.75 H. P. and a pitch line speed of 660 feet per minute

$$P = \frac{18.75 \times 33{,}000}{660} = 940 \text{ pounds approximately.}$$

The width of gear is given by the formula:

$$w = \frac{P}{f \times C_1 \times C_2} \text{ where}$$

w = width of tooth on gear in inches.

P = pressure on tooth in pounds.

f = permissible fiber stress in thousands of pounds per square inch.

C_1 = coefficient depending on diametral pitch and number of teeth on gear. Its values can be obtained from table given as Fig. 2.

C_2 = coefficient depending on pitch line speed. Its values can be obtained from curve in Fig. 3. If we suppose a gear of steel we may use a fiber stress of 8,500 pounds per square inch. We therefore get as the width of tooth:

$$w = \frac{940}{8.5 \times 52 \times .5} = 4\tfrac{1}{4} \text{ inches, approximately.}$$

Besides the weight of the rotating part and the pressure on the gear we must, when figuring the shaft and bearings, take

into consideration the unbalanced magnetic pull caused by a displacement of the armature of the motor in relation to the poles. If B is the density given in lines of force per square inch at air gap, A is the area of pole face in square inches, and k is a constant which

for 4-pole machines $= 2$
6-pole machines $= 4.7$
8-pole machines $= 7$

Then the magnetic pull per pole $= \dfrac{B^2 \times A}{k \times 72{,}134{,}000}$ pounds.

This formula gives us in our example a magnetic pull per pole $= \dfrac{58{,}800^2 \times 29}{2 \times 72{,}134{,}000} = 700$ pounds approximately. The pull per inch of the circumference of pole bore

$$= \frac{4 \times \text{pull per pole}}{\pi \times \text{pole bore}} = \frac{4 \times 700}{\pi \times 9.69} = 92 \text{ pounds approximately.}$$

If we now suppose a displacement of armature of 25 per cent of the normal air gap, the ratio between air gap and displacement $= 4$. In the diagram, Fig. 4, reading on vertical side the pull per inch of the circumference $= 92$, and on the horizontal side the ratio between air gap and displacement $= 4$. The curve 55 passing through the intersection point of 92 and 4 indicates that the half value of the maximum magnetic pull per inch of circumference of pole bore is:

$\dfrac{M\,max}{2} = 55$ pounds. Thus $M\,max = 55 \times 2 = 110$ pounds. In order to give the values of $\dfrac{M\,max}{2}$ as exactly as possible for a wide range of values of magnetic pull and displacement ratios, the diagram in Fig. 4 contains two sets of curves, one in the lower right corner on a small scale, and one in the upper left corner on a larger scale.

The total magnetic pull on armature

$$= \frac{4 \times \text{radius of armature} \times M\,max}{\pi}$$

In our example the radius of armature is 4.75 inches and the $M\,max$ is 110 pounds. The total magnetic pull therefore

$$= \frac{4 \times 4.75 \times 110}{\pi} = 665 \text{ pounds.}$$

When the unbalanced magnetic pull is acting in the same direction as the weight of the rotating part the shaft is subjected to its worst strain. Therefore by adding those two forces we get the resulting force

$$W = 665 + 150 = 815 \text{ pounds.}$$

In general the location of this force W on the shaft will practically lie on the center line of the armature.

The forces R_1 and R_2 acting on the bearings, as shown in Fig. 1, will be found from the following equations:

$$R_1 = \frac{W \times b + P \times (c + L)}{L} = 1{,}680 \text{ pounds}$$

$$R_2 = W + P - R_1 = 75 \text{ pounds}$$

Diameter of Shaft.

The diameter of shaft at W, in Fig. 1, can now be figured. The bending moment at W is:

$$M_b = R_1 \times a = 13{,}650 \text{ inch-pounds.}$$

The twisting moment for 18.75 H. P. and 700 R. P. M. is:

$$M_t = \frac{18.75}{700} \times 63{,}024 = 1690.$$

The combined moment M_c of M_b and M_t is:

If M_b is greater than M_t

$$M_c = .975 \times M_b + .25 \times M_t;$$

or if M_b is less than M_t

$$M_c = .6 \times M_b + .6 \times M_t.$$

VALUES OF C_1 FOR 15° INVOLUTE TEETH OF ONE INCH FACE WHICH PRODUCE FIBRE STRESS OF 1000 POUNDS PER SQUARE INCH.

Number of Teeth in Gear.	Circular Pitch. 3.14	2.52	2.10	1.80	1.57	1.40	1.25	1.14	1.05	0.98	0.89	0.84	0.79	0.70	0.63	0.57	0.52
	Diametral Pitch. 1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4½	5	5½	6
12	210	168	140	120	105	93	84	76	70	65	60	56	52	47	42	38	35
13	220	178	147	126	110	98	88	80	73	68	63	59	55	49	44	40	37
14	226	180	151	129	113	100	90	82	75	69	65	60	56	50	45	41	38
15	236	189	157	135	118	105	94	86	79	73	67	63	59	52	47	43	39
16	242	194	161	138	121	107	97	88	81	74	69	64	60	54	48	44	40
17	251	200	167	143	125	112	100	91	84	77	72	67	63	56	50	46	42
18	261	208	174	149	130	116	104	95	87	80	75	70	65	58	52	47	43
19	273	218	182	156	136	121	109	100	91	84	78	73	68	61	54	50	45
20	283	227	189	162	142	126	113	103	94	87	81	75	71	63	57	51	47
21	289	232	193	165	144	128	116	105	96	89	83	77	72	64	58	52	48
23	295	236	197	169	147	131	118	107	98	91	84	79	74	65	59	53	49
25	305	245	203	174	152	136	122	111	102	94	87	81	76	68	61	55	51
27	314	250	210	180	157	140	126	114	104	97	90	84	78	70	63	57	52
30	320	256	213	183	160	142	128	116	106	99	91	85	80	71	64	58	53
34	327	262	218	188	164	146	131	119	109	101	94	88	82	73	66	59	54
38	336	263	224	192	168	148	134	122	112	103	96	90	84	75	67	61	56
43	346	277	230	198	173	154	138	126	115	106	99	92	86	77	69	63	57
50	352	282	235	200	176	156	140	128	117	108	100	94	88	78	70	64	58
60	358	286	238	204	179	159	143	130	119	110	102	95	89	80	71	65	59
75	364	292	243	208	182	162	146	132	121	112	104	97	91	81	73	66	61
100	371	297	247	212	185	164	148	135	124	114	106	99	93	82	74	67	62
150	377	302	251	215	188	167	151	137	126	116	108	100	94	84	75	68	63
300	384	308	256	219	192	171	152	140	128	118	110	102	96	85	77	70	64
Rack	390	312	260	223	195	173	156	142	130	120	112	104	97	87	78	71	65

Fig. 2.

In our example where $M_b > M_t$, $M_c = .975 \times 13{,}650 + 25 \times 1690 = 13{,}720$, approximately.

Now, if f = fiber stress in shaft per square inch, and D = diameter of shaft at W in inches, the moment of resistance

$$M_r = .1 \times f \times D^3$$

Therefore if we put $M_r = M_c$,

$$D = \sqrt[3]{\frac{M_c \times 10}{f}}$$

If we suppose a fiber stress $f = 8{,}500$ pounds per square inch, we get

$$D = 2\tfrac{1}{2} \text{ inches approximately.}$$

The minimum diameter of shaft in the bearings can be figured in the same manner; for instance at R_1 we get

$$M_b = 5{,}050$$
$$M_t = 1{,}690$$
$$M_c = 5{,}340$$

which give us a minimum diameter at $R_1 = 1\ 13/16$ inches, approximately. It is of course an advantage, however, to have the diameters of the journals somewhat larger than calculated. In this case the mechanical conditions may give a diameter at $R_1 = 2\tfrac{1}{4}$ inches.

Calculation of Journals.

In the bearings it is not advisable to exceed a pressure per square inch of 150 pounds, nor should the product of this pressure by the peripheral velocity in feet per minute in the

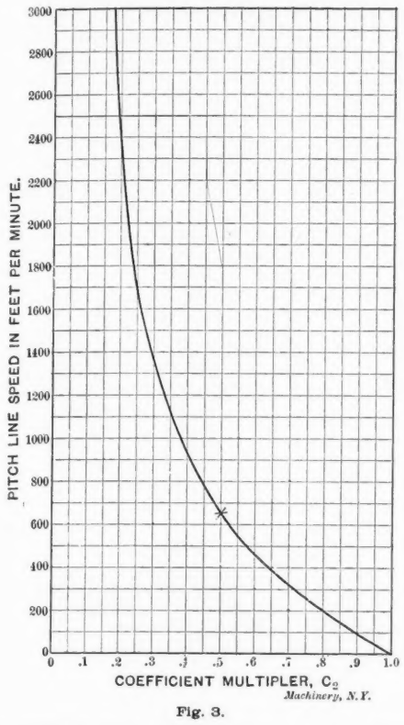


Fig. 3.

journal be greater than 55,000, when grease lubrication is used. For oil lubrication this product can be somewhat higher. If in our example we assume a pressure of 130 pounds per

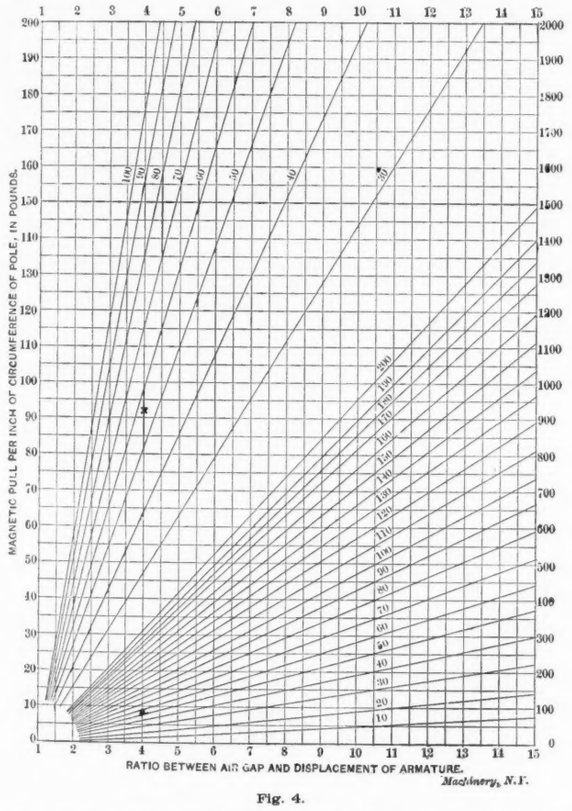

MAGNETIC PULL PER INCH OF CIRCUMFERENCE OF POLE, IN POUNDS.
RATIO BETWEEN AIR GAP AND DISPLACEMENT OF ARMATURE.
Machinery, N.Y.

Fig. 4.

square inch, with the diameter of shaft at $R_1 = 2\frac{1}{4}$ inches, we obtain length of journal $= \frac{1680}{130 \times 2.25} = 5\frac{3}{4}$ inches. At 700 R. P. M. and diameter of shaft $= 2\frac{1}{4}$ inches, and a pressure of 130 pounds per square inch in journal, the product of pressure by velocity will be $= \frac{130 \times \pi \times 2.25 \times 700}{12} = 53{,}500$, approximately.

Maximum Deflection of Shaft.

For calculating the maximum deflection S of the shaft we have the following formula:

$$S = \frac{W \times a \times b\,(2L - a)}{9 \times E \times I \times L} \sqrt{\frac{a\,(2L - a)}{3}}$$

where S is in inches and

W = the resulting force acting on the shaft in pounds,

L = distance between centers of journals in inches,

a = shortest distance between center line of one bearing and the acting point of force W,

$b = L - a$ in inches,

E = Modulus of elasticity,

= 29,000,000 for steel,

= 27,000,000 for wrought iron,

I = moment of inertia of shaft, $= .0491 \times D^4$, where D is the diameter of shaft in inches.

In this example we get the maximum deflection,

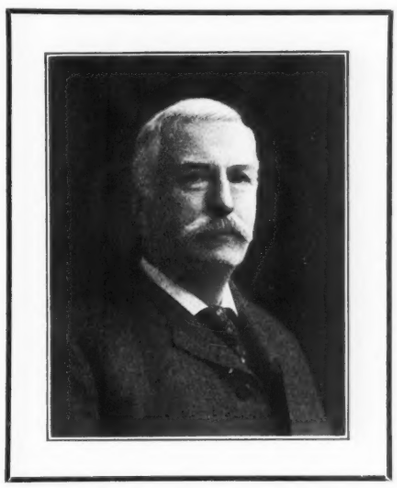
$S = .0027$ inches approximately.

Most of the formulas given above are empirical, and give only approximate results, but they are exact enough for practical use.

* * *

DEATH OF MR. GEORGE A. GRAY.

Mr. George A. Gray, the founder and for many years president of the G. A. Gray Company, Cincinnati, O., died suddenly at his home in that city, on June 14th, of heart trouble. Mr. Gray was born in 1839 on a farm in Illinois and his boy-



hood was spent there. Later he came to Cincinnati and served an apprenticeship with Miles Greenwood, devoting his attention to the study of mechanics; after which, with Messrs. Gordon & Gaff, he bought the machine tool department of the old Niles Works which had been started by a man named Niles on Front Street, east of Butler, calling the new firm Gaff, Gray & Gordon. After continuing at that location for several years the plant was removed to Hamilton, Ohio, and the name changed to Niles Tool Works. Later Mr. Gray sold his interest in the firm and started the manufacture of planers in Cincinnati, in which business he continued until last May, when he retired from the active management of the company which bears his name.

Mr. Gray's mechanical experience covered a wide field. As a young man during the Civil War he was engaged in building monitors for the United States Government, and about the same time he invented a multiple firing gun, the first of its kind that was ever made. He was universally regarded as the foremost mechanic in his line.

Mr. Gray was a man of singularly attractive character, quiet and retiring in his ways, but a good friend and upright in all his business dealings. His death will cause great sorrow to all those with whom he was associated by ties of friendship or business.

LOAD ON A JIB CRANE.

DYER SMITH.

A reader of MACHINERY has propounded a very interesting problem in connection with the strength of a jib crane. He states that the column of the crane, as shown in Fig. 1, is cast iron, has all the appearance of being sound, and is supposed to have ¾ inch of metal. The dimensions are as per sketch. The compression member consists of two 7-inch channels, weighing 11¼ pounds per foot, arranged back to back with a 3-inch space between them for the trolley to operate in. These are fastened together at each end, and the outer ends are supported by two 1-inch rods. The question raised is whether it will be safe to suspend 4 tons from the end of the 11-foot jib.

Calling a ton 2,000 pounds, the force conditions, reduced to simplest terms, will be as shown in Fig. 2. A compound beam with compressive stress, as indicated in the lower part of Fig. 2, would evidently be an equivalent case. Considering the column as a compound beam, the moment diagram will be as shown in Fig. 3. From this it will be seen that the

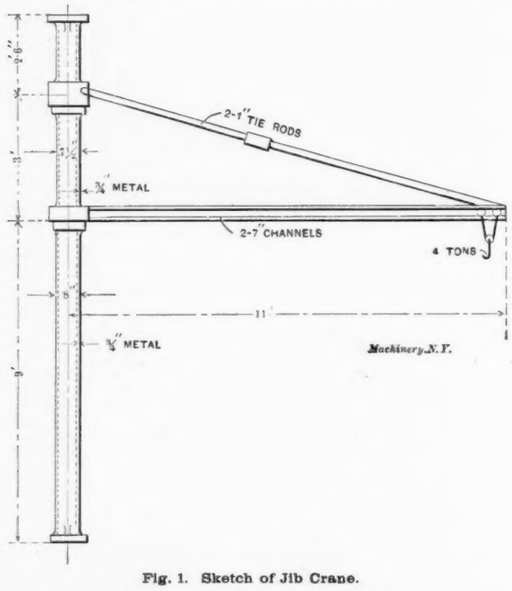


Fig. 1. Sketch of Jib Crane.

maximum bending moment on the column equals 54,614 foot-pounds, exerted in the axis of the jib, or 5½ feet from the upper end of the column.

To find the maximum fiber unit stress for the case of a beam subject to flexure by transverse loads and also to compression in the direction of its length, we find in Merriman's Mechanics of Materials, 9th edition, page 267, the formula

$$S_1 = \frac{M c}{I - \frac{n P l^2}{m E}}$$

where S_1 = maximum fiber unit stress,
M = maximum bending moment in pound-inches,
c = distance from the neutral axis to the remotest fiber,
I = moment of inertia of the cross section,
P = longitudinal compressive force = 8,000 pounds,
E = coefficient of elasticity = 15,000,000 pounds per square inch for cast iron,
n-m = numbers depending on arrangment of ends and kind of loading,
l = length of span of the beam, in inches.

In the above, M, the maximum bending moment, = 54,614 foot-pounds = 54,614 × 12 inch-pounds, and c = 3¾ inches.

I, for hollow column, = .0491 $(d^4 - d_1^4)$, where d and d_1 are the external and internal diameters, 7½ and 6 inches, and hence I = 91.5. The approximate value of $\frac{n}{m}$ is 1-12. The span l, in the equivalent case of the compound beam, is closely the distance between supports, or in our case, 144 inches. Hence we have

$$S_1 = \frac{54{,}614 \times 12 \times 3\frac{3}{4}}{91.5 - \frac{1}{12}\,\frac{8{,}000 \times 144 \times 144}{15{,}000{,}000}}$$

= 27,150 pounds per square inch.

To this should be added $\frac{P}{A}$, where A is the cross sectional area of the column, for the maximum compressive unit stress.

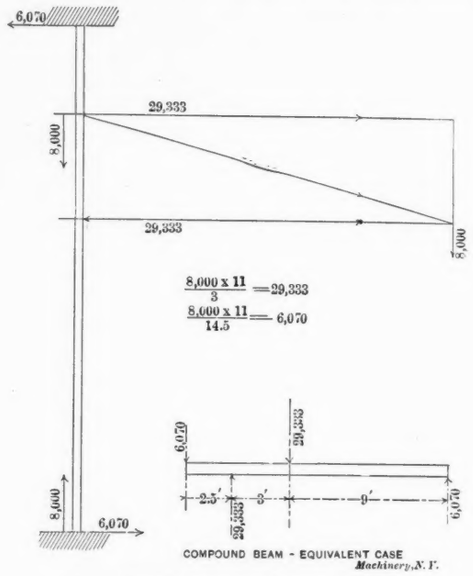


Fig. 2. Forces Acting on Column of Crane.

A = 15.9 inches, making $\frac{P}{A} = \frac{8{,}000}{15.9} = 503$, and hence $S = S_1 + \frac{P}{A}$ = 27,650 pounds per square inch, for compression.

For tension, $S = S_1 - \frac{P}{A}$ = 26,650 pounds per square inch.

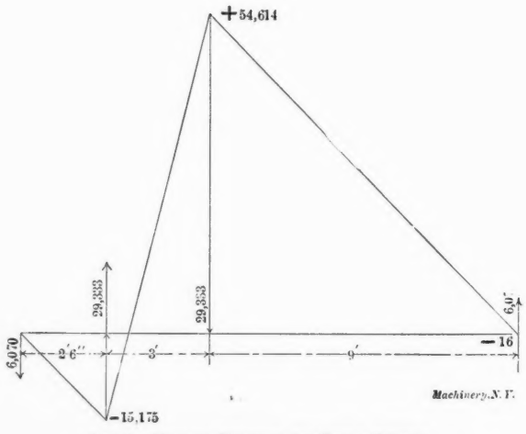


Fig. 3. Moment Diagram for Crane Column.

Since the average ultimate strength of cast iron in tension is 20,000 pounds per square inch, it follows that the column will probably fail when a load of 8,000 pounds is lifted at the end of the jib.

The above method applies also to the discussion of the channels which are under combined flexure and compression. The slenderness-ratio of this column is too large for good engineering practice, and entirely insufficient for a load of 4 tons.

* * *

Copper-vanadium is said to be possessed of great resistance and hardness. Copper-vanadium alloys in different proportions are recommended as being suitable for the manufacture of guns, propeller blades, valves, steam turbine blades, and for many other uses for which a hard copper alloy possessing great tensile strength is required.

ESTIMATING THE COST OF MACHINE WORK.

J. A. WEBSTER.

J. A. Webster.

I have seen numerous articles, both in MACHINERY and other technical periodicals, relating to estimating in the abstract, but very rarely anything in the nature of a concrete example; and while the articles have generally been good and interesting reading, it seems to me that to take up a machine in detail, and analyze the various operations to be performed, will be of equal or greater value to the young machinist.

For instance, there has lately been placed on the market a pencil sharpening machine which presents some operations that appear at first, if not somewhat difficult, at least not easy to perform economically. We were requested to estimate on making a set of tools suitable for turning out the parts so they would be interchangeable, and also for making the first 1,000 machines.

There are some 24 parts in the machine which are shown in detail in the figures throughout the text, numbered to correspond with the part numbers, and a sectional view of an assembled machine is shown in Fig. 1.

In making up the estimate the following order was adopted: First, the amount of material required for the various parts was calculated, then each part was taken up, methods of machining considered and discussed, the time required for the different operations estimated and tabulated, and lastly the cost of the tools required was made up. The experience of the writer is that it is best to leave the tools to the last for the reason that it is generally necessary to analyze the operations in detail in order to decide what tools must be made, and while doing this it is most convenient to tabulate the time required to do the work.

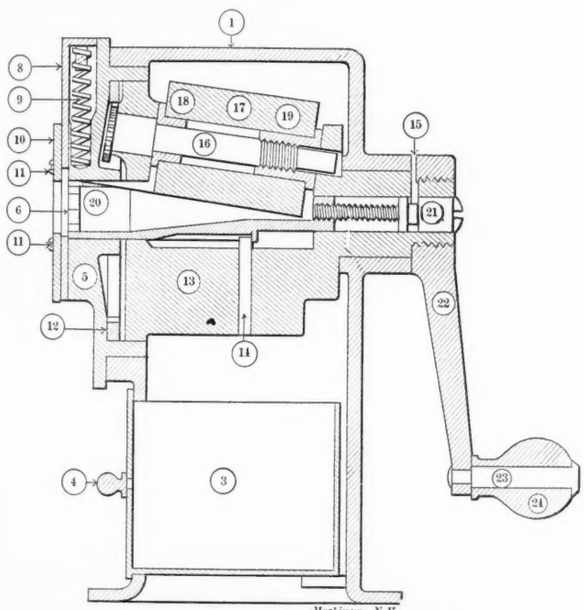


Fig. 1. Sectional View of an Assembled Machine.

We were furnished a working model that was substantially accurate in all its details, and the weights of the cast iron parts were not calculated, but obtained by weighing those of the sample and adding 20 per cent. for loss in machining, broken and spoiled parts, etc. While this is sufficiently accurate for small articles, and saves time, on heavy castings it is best to calculate the weights as accurately as possible.

The cast iron parts of the sample weighed 2½ pounds. Adding 20 per cent., as above, gives 3 pounds as the weight for one set. As the castings are light, and to facilitate machining must be of the very best quality of soft gray iron, and part marked No. 1 is cored work, the price was set at 6½ cents per pound. The price at which the order was afterward placed was 7 cents and the average weight 2.88 pounds.

The weight of the other materials was all calculated, and the whole tabulated as below.

Cast Iron.

Parts 1, 5, 13, and 22.....3 lb. per set, 3,000 lb. at 6½c., $195

Bessemer and C. R. Rods.

Part 2, 3-16 in. dia.....	50 feet	5 lb.
Parts 7, 14, and 15.....	150 feet	4 lb.
Part 16, ½ in. dia.....	200 feet	131 lb.
Parts 4 and 21, 5-16 in. dia..	110 feet	28 lb.
Total..	168 lb. at 5c.....	8.40

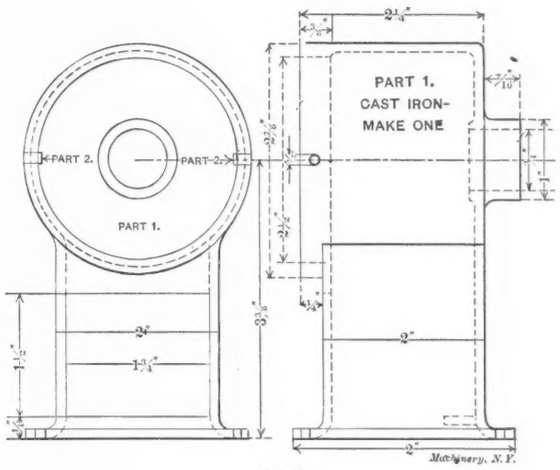


Fig. 2.

Flat C. R. Steel.

Part 6, ⅛ x ⅜ in.....	167 lin ft. 26 lb. at 8c.....	2.08
Part 3, 1-16 in. thick.....	24 sq. ft.	
Part 8, 1-16 in. thick.....	20 sq. ft.	
Part 10, 1-16 in. thick.....	18 sq. ft.	
Total	62 sq. ft. = 161 lb.	
Part 3, 1-32 in. thick.....	121 sq. ft. = 157 lb.	
Total....	318 lb. at 5c..	15.90

Tool Steel—Annealed.

Part 17, 1 1-16 in. dia.....125 feet 377 lb. at 16c..... 60.32

Piano Wire.

Part 9.....1 lb..... .50

Round Brass.

Part 18, ½ in. dia.....	27 feet.	
Part 19, ½ in. dia.....	55 feet.	
Part 20, ½ in. dia.....	212 feet.	
Total....	294 feet = 209 lb. at 16c.	$33.45

Half Hard Sheet Brass.

Part 12, 1-16 in. thick.....80 sq ft. = 220 lb. at 15c. 33.00

	66.45	
Less 100 lbs. scrap at 7c.	7.00	
		59.45
Rivets for part 3.....		.50
Part 11, 3 x 48 x ⅛ R. H. Iron Mch. Screws, 27 gross at 15c.....		4.05
Part 24, Hard Maple, 1,000 pieces.....		3.50
Total		$349.70

In addition to the above it was required to supply and pack with each machine four ⅝-inch No. 7 R. H. wood screws. Each machine must be wrapped with paper and packed in a locked-corner, wooden box and these packed two dozen in a case and delivered f.o.b. cars. So we have:

27 gross screws at 15c.....	$4.05
1,000 individual boxes.....	15.00
42 cases at 30c.....	12.60
	$31.65

Parts 1, 5 and 22 were required to be enameled in black, two-coat work, and parts 8 and 10 and the front plate of part 3 nickel plated and polished. As we did not have facilities for

JOHN A. WEBSTER was born at Berrien Springs, Mich., 1852. He received a business education and practiced architecture in New York for ten years, but finally found that machine work was more to his taste. He has served as designer and superintendent and in various other positions of trust for several concerns in New York City. His specialty is devising means and tools for difficult work in metal.

doing this, they would have to be sent out, so it was necessary to add the cost of cartage back and forth.

Enameling parts 1, 5 and 22, 1,000 sets at 6c..	$60.00
Nickel plating parts 8, 10 and front of part 3, 1,000 sets at $2.65 per 100.....	26.50
Cartage	1.50
	$88.00

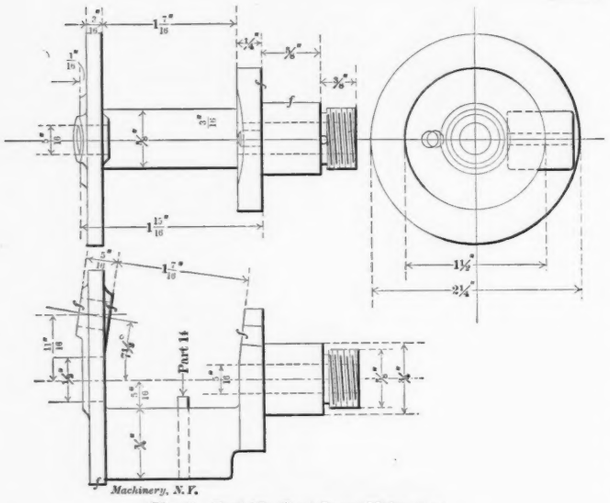
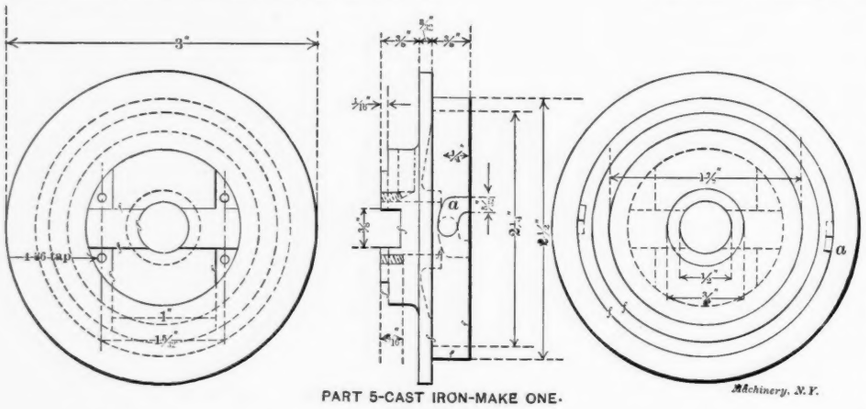


Fig. 3. Part 13—Cast Iron—Make one.

This brings the total cost of material, packing and work required to be sent out to $469.35, to which we thought wise to add to cover freight, express and cartage on materials and finished goods $10.65, making a total of $480.00.



PART 5-CAST IRON-MAKE ONE.

Fig. 4.

Coming to the estimating of labor, we listed it under three classes of employees, viz., toolmakers, journeymen and boys, and calculated the time required for the whole lot of 1,000 machines.

The first item considered was the cleaning and snagging of castings. Part 1 being a cored piece, it was expected that there would be considerable work on this, but the other parts could be tumbled in the foundry and would come to us clean and in good condition to put on the machines. The work of cleaning all the castings appears in the first item in table headed Labor.

Taking up part one, we concluded to put it on a monitor lathe for the first operation, holding it in a two-jawed chuck, the jaws of which were made to fit the casting. Here we would drill and ream the hole in the hub, face the inside boss, and follow by boring and facing the opposite large end of the casting. For this boring and facing operation two tools would be required, one for the roughing and another for the finishing cuts. The next operation is facing the outer end of the hub, which would be done on a drill press, using a counterbore and running down to a stop. The casting could now be placed in a jig, accurately located by the finished surfaces and the holes, seven in number, drilled, which completes the machine work on this part.

For the operations on the monitor lathe, we allowed time as follows: Chucking, ½ minute; drilling and reaming, ½ minute; rough boring, 1¼ minutes; finish boring, ¾ minute; total, 3 minutes, or 20 pieces per hour. We worked 9 hours per day, but to cover time lost in grinding tools and incidental delays that almost always occur, it is our custom to figure on only 8 hours actual work. This would give 160 pieces finished each day, or 6¼ days for the lot. For good measure we put it down in the table as 7 days. Operations of this kind require more or less of the foreman's time in adjusting tools, etc., and to cover this it is our practice to add a percentage of the time, placing it in the column marked "toolmaker." In this case we allow ten per cent., or 6 hours for even figures.

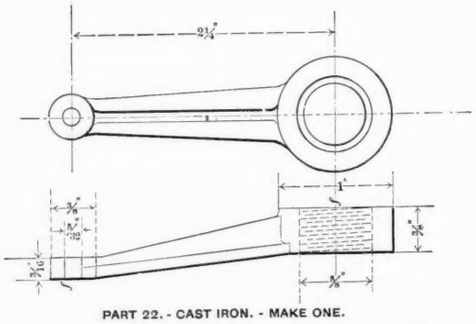
The counterboring of the hub will go quite rapidly, but as much time will be required to handle and move the pieces as to do the work. Still, the whole lot should be handled in 1½ days, and the job will not require so much of the foreman's time, as before, so we will put only one hour in the "Toolmaker" column.

In the final operation of drilling, we will allow ½ minute to place the piece in the jig and remove it and 1½ minutes to do the drilling, which gives 30 per hour, or 240 per day. For good measure we will say 4½ days for the lot, with 3 hours in the toolmaker column.

Next we will take up part 5. As before, the first operation is on the monitor lathe, and as the piece has a good-sized round hub, it can be held in a three-jaw chuck, though it will probably be better to use the two-jaw and make special jaws to fit the castings. At this operation we will drill the ½-inch hole, bore and face to receive the internal gear, and turn down on the outside and shoulder to fit in Part 1. As before we will use roughing and finishing tools, and for the last sizing cut will use a tool in the slide rest in order to secure greater accuracy.

For the chucking we will allow ½ minute; drilling and reaming, ½ minute; boring and facing for gear, rough and finish, each one minute; turn and shoulder, 1 minute; and sizing finish, ½ minute, making a total of 4½ minutes, or about 13 pieces per hour. Say 100 pieces per day. This will require 10 days for the lot and we will put 1 day in the Toolmaker column, as the foreman will have to watch this work rather closely

Next we will have two milling operations which we will calculate to have going on at the same time on two adjoining millers, so that one boy can attend both machines, changing the pieces on one while the other is running. This faces the hub and cuts the slots for parts 6 and 8. To insure accuracy in the width of these slots, we will use interlocking cutters, and special jaws will be required for the miller vises.



PART 22. - CAST IRON. - MAKE ONE.

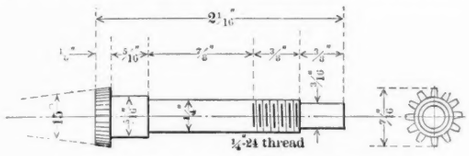
Fig. 5.

In fixing the time, we again take each movement in detail, but it is not considered necessary to set down the time for each here, as it would make this article very lengthy and probably wearisome. The time decided on as necessary for the different operations will be found in Table No. 1.

The third operation will be drilling the holes for screws and spring, and will be done in a jig, after which the screw holes

will be tapped. The fifth and final operation is milling the two bayonet slots (*a a*) that engage the two pins in part one. It is necessary that these be very accurately located and of the same profile, so that the shoulder will come to a bearing on both sides at the same time. This was a matter to which we gave considerable thought and finally decided to make a special machine to do the work. The reasons for this decision were that our millers did not run at a high enough speed for such small mills (5-32 inch diameter), that to speed up a machine and make a fixture would cost about two-thirds as much as a complete machine, and finally that we could make the special machine cut both slots at the same time. (A description of the machine that was afterward made for this work was published on page 492 of the May, 1904, issue of MACHINERY.)

Taking up next Part 13, we find a piece that will require very careful handling all through, for if this part is not correct, there will be trouble when the machine is assembled. This

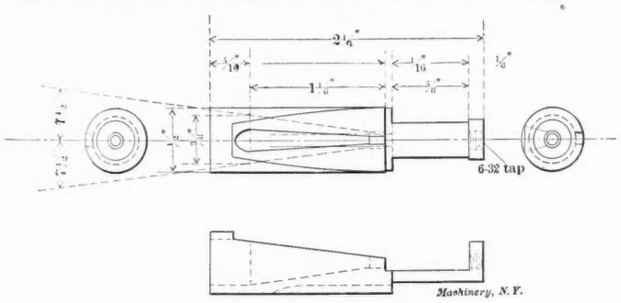


PART 16-MCH STEEL-MAKE ONE.
PINION-P.D. ¾" O.D. 7/16" PITCH 32-12 TEETH.

Fig. 6.

part revolves on two bearings, one of which is the shaft end to which the crank is attached, and the other the projecting end of part 20, which fits in the central holes, so that these holes must be in line and the shaft concentric and in line therewith. The two holes for the bearings of part 16 must be accurately located with relation to the axis on which the part revolves, all of which leads to the conclusion that the proper place to begin the work is with the central holes through the part.

For the first operation we will chuck the piece in the monitor lathe, by the hub end, and drill the ½-inch hole through the larger disk and the 5-16-inch hole through the hub end, reaming both holes with a special reamer to insure their being in line. For the second operation we will put the piece on a mandrel, turn and face the hub end, and size the large disk. Cutting the screw threads and facing the boss on the large disk will constitute the third and fourth operations. The fifth operation will be milling the disks to form the bearings for



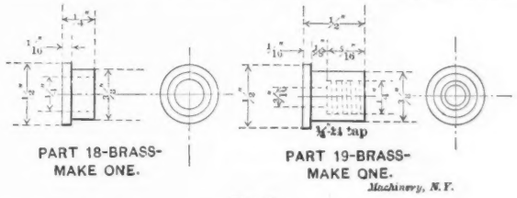
PART 20-BRASS-MAKE ONE.

Fig. 7.

the shoulder of part 16 and the bushings 18 and 19, and for this we will make a special holding fixture to insure the cuts being made in proper relation to the central holes. Drilling the holes in a jig will form the sixth operation, and to insure accuracy we will ream the holes for the cutter shaft (part 16) in a jig, making seven operations in all. In all the fixtures used, the work will be located by a pin or mandrel passing through the central holes, and by this means we should be able to produce parts that are exact duplicates.

Next we will take up Part 22, which is quite simple. We will hold it in a chuck with special jaws on the monitor, drill, counterbore and tap the large hole and then drill and counterbore the small hole in a jig on the drill press.

Taking up next the round-rod parts, we find two—Parts 16 and 20— that will require to be made with great care. Considering first, Part 16, it is apparent that the bearings upon which it turns and the parts of the shaft upon which the parts 18 and 19 fit, must all be in line, concentric, and of accurate dimensions, otherwise the cutter will run out of true and cut upon only one side. Such work being almost impossible to produce on the screw machine, we will calculate on finishing the piece with the exception of that part which fits in part 18, which we will leave slightly over size. To complete the shaft, we will make a special chuck and steady rest for the precision bench lathe and finish the shaft together with the shoulder, on this machine. While this makes an additional

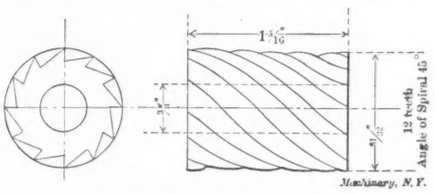


PART 18-BRASS-MAKE ONE.
PART 19-BRASS-MAKE ONE.

Fig. 8.

operation and adds to the cost of tools, it will save time in assembling, a thing that should never be lost sight of. There yet remains to cut the teeth of the pinion, which is accomplished by setting it at the proper angle and treating as if it were a simple spur gear.

Analyzing part 20, it is seen that the cone-shaped hole should be concentric with the outside, otherwise the pencil being supported centrally by the holding chuck would be given a wabbling movement as the machine was turned, and there would be liability that the point of the lead would be broken. To drill and ream this hole concentric with the outside would hardly be practicable, so we will use stock 1-32-inch over size and finish the pieces on a mandrel in the precision lathe. After



PART 17-TOOL STEEL-MAKE ONE.

Fig. 9.

this there will be two milling operations to finish the part.

Parts 18 and 19 will also need to be made with considerable care, but present no special difficulties. The holes should be true with the outside and the thickness of the flanges should not vary. The holes must be left somewhat undersize so they may be reamed after being forced into the cutter.

The remaining round parts, 2, 4, 7, 14, 15, 21 and 23, are quite simple, and will need no special mention.

Next we will consider Part 17. We have a universal mill-

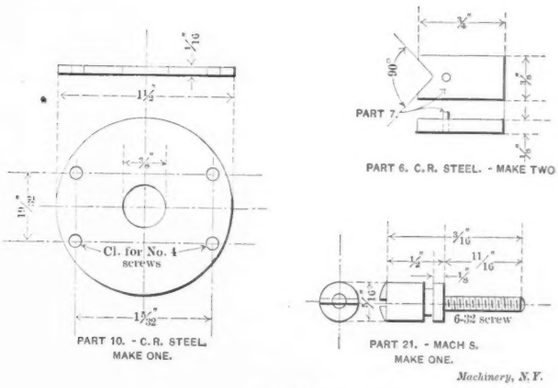


Fig. 10.

ing machine upon which we can handle pieces 18 inches in length, so we will cut the bars that length, centre, and turn down to finish size. In order that we can mill the full length, we will put a ⅜-inch hole in one end 1 5-16 inch deep and in this we will force a short piece of ⅜-inch drill rod, which we can catch in the split chuck of the indexing fixture. We will make a special indexing fixture and provide a rest for the bar under the cutter. We will also make a

special cutter of high-speed steel. With this we should be able to mill about 2½ inches per minute. Making allowance for indexing, changing and grinding cutter, we can count on averaging 125 inches per hour. The length of a 45-degree spiral on an 18-inch bar is 25½ inches, and as there are 12 teeth to cut, we will have 306 inches to cut on each bar. One bar will make 12 parts. So we will require 83 1-3 bars for 1,000 parts. As some will probably be lost in hardening, we will mill 87 bars, which, multiplied by 306, gives 26,622 inches as the total length to be milled. Dividing by 125, the number of inches milled in one hour, we have 213 hours, or 26 days 5 hours. For good measure we will say 28 days.

The next operation will be to remove the burr raised by the cutter, and to do this we will make a hollow mill which can be run over the bars. Next we will drill, ream and cut to length, when the parts will be ready for hardening. After they are hardened the bushings must be forced in, which can be done very readily with a foot press. When both bushings are in place they must be reamed with a special hand reamer, which makes all ready for the final operation of grinding the teeth.

Taking up, now, part 6, we will cut to length with a shearing die, mill the end as shown and drill the hole for the small pin in a jig.

The remaining parts with the exception of the helical spring which is wound on a mandrel and cut to length, are of sheet metal, and taking up No. 3, we find it composed of two pieces. The front consists of a flat piece of steel 1-16 inch thick, pierced with five holes and having stamped on its surface the name and address of the manufacturers, etc. As no great accuracy is required here, we will make a combined blanking and piercing die. For the second part we will require a blanking and two forming dies. Part 8 will require a blanking, two piercing and one forming dies; part 10 will need only one combined blanking and piercing die.

Part 12 is composed of two similar pieces assembled together. We will first blank out in rings and afterward cut the teeth at a single and final operation. To secure a good result these dies must be made with the greatest accuracy, and the work very carefully handled.

Having now covered the making of all of the parts, we will take up the assembling and tabulate the time required for this in Table No. 2. Beginning with part 1 we have only to put in the two pins that engage in the bayonet slots of part 5.

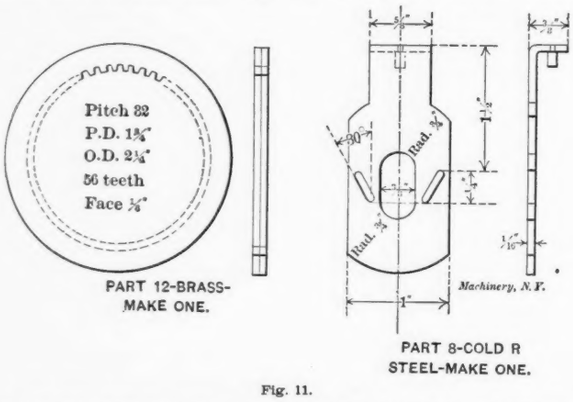


Fig. 11.

As the castings will vary somewhat in thickness, we have cut the pins a little long and will make a fixture for the foot press that will enable us to force them in against a stop which will leave the inner end projecting to the proper height. The outer end will then be filed off flush with the surface of the casting.

Taking next part 5 we will force the internal gear No. 12 into place. This work we will calculate to do on the power press, having made fixtures to accurately locate and guide the gears. We will hope to make our parts so exact that friction will hold the gears in place, otherwise they will have to be pinned to prevent turning. We are now ready to assemble Parts 5, 6, 8, 9 and 10. The putting together of these parts is so plainly shown in the drawings that it is not necessary to describe it in detail.

Part 13 is next in order. We will first force pin 14 in place, making provision in our means for doing this to insure the end projecting to the proper height. Then we will put Part 21 in place and drive in retaining pin No. 15, being careful that it does not come down hard on the screw and prevent its turning. We can now slip in Part 20 and turn screw 21 until it is drawn into place. Next we will press the two bushings 18 and 19 into the cutter 17. In case we have not done our work perfectly and the lengths when assembled vary we shall have to counterbore the bushing at one end, preferably No. 18, to bring them all to the same length.

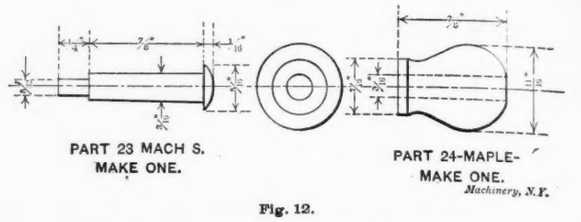


Fig. 12.

We have now remaining only the riveting together of No. 3, and the assembling and riveting together of Nos. 22, 23 and 24, when the machine is ready to put together, and the design is such that this is readily done without the use of tools.

TABLE NO. 1.

Labor: Cleaning Castings and Machining Parts.

Part No.	Operation.	Toolmaker. Days.	Hrs.	Journeyman. Days.	Hrs.	Boy. Days.	Hrs.
1-5-13 and 2	Snagging and cleaning castings					4	0
1	Drilling and boring—monitor.	0	6			7	0
	Counterboring—drill press.		1			1	4
	Drilling—jig—drill press.		3			4	4
		1	2			13	0
5	Drilling and boring—monitor.	1	0			10	0
	Milling chuck part slats—2 milling machines		5			6	0
	Drilling—jig—drill press.		2			4	0
	Milling bayonet slots—special milling machine		3			3	4
		2	2			23	4
13	Drilling and reaming—monitor		3			4	0
	Turning and facing—engine lathe		1	5	0		
	Cutting screw threads—monitor lathe opening dies.		2			3	0
	Facing boss—drill press—counterbore		1			2	0
	Milling bearings—milling machine		2			3	4
	Drilling—jig—drill press.		1			4	0
	Reaming—jig—drill press			2	0		
		1	2	7	0	16	4
22	Drill counterbore and tap—monitor		2			4	0
	Drill and counterbore—drill press					1	0
			2			5	0
16	Forming—screw machine.		3			4	0
	Turning—precision lathe			4	0		
	Cutting teeth in pinion—milling machine		5			10	
		1	0	4	0	14	0
20	Forming—screw machine		3			4	0
	Turning—precision lathe			2	0		
	Milling—1st operation—milling machine		2			2	0
	Milling—2d operation		1			1	0
	Burring					2	0
			6	2	0	9	0
18	Forming—screw machine.		2			1	4
19	Forming—screw machine.		3			3	0
2	Forming—screw machine.		2			1	0
4	Forming—screw machine.		3			2	0
7	Forming—screw machine.		2			1	0
14	Forming—screw machine.		1				5
15	Forming—screw machine.		1				4
23	Forming—screw machine.		3			2	5
		1	4			7	6
21	Forming—screw machine.		4			3	0
	Slotting—hand miller.		1			1	0
			5			4	0
17	Cutting to length—power hack saw					1	0
	Centering and drilling engine lathe			2	0		
	Turning engine lathe			1	4		
	Cutting drill rod—screw machine		1			3	0
	Milling teeth—universal miller	2	0			28	0
	Burring						4

Part No.	Operation.	Toolmaker. Days.	Hrs.	Journeyman. Days.	Hrs.	Boy. Days.	Hrs.
	Drill, ream and cut off—monitor			7	0		
	Face to length—drill press counterbore					3	0
	Hardening	4	0			4	0
	Grinding	2	0			10	0
		8	1	10	0	49	4
6	Cutting to length—power press		1				3
	Milling—milling machine		2			2	0
	Drilling jig—drill press		1			1	0
			4			3	3
9	Winding—speed lathe		1				4
	Cutting to length—special shear						2
			1				6
3	Blanking and piercing front—power press		1				2
	Stamping—power press		1				2
	Blanking second part—power press		1				3
	Forming — 1st operation — power press		1				3
	Forming — 2d operation — power press		1				3
			5			1	5
8	Blanking—power press		1				2
	Piercing — 1st operation — power press		1				3
	Piercing — 2d operation — power press		1				3
	Forming—power press		1				3
			4			1	3
10	Blanking and piercing—power press		1				3
12	Blanking—power press		2			1	0
	Cutting teeth—power press		2	1	4		
			4	1	4	1	0

SUMMARY.

	Cleaning castings					4	0
1	3 operations	1	2			13	0
5	4 operations	2	2			23	4
13	7 operations	1	2	7	0	16	4
22	2 operations		2			5	0
16	3 operations	1	0	4	0	14	0
20	5 operations	6	0	2	0	9	0
18	1 operation		2			1	4
19	1 operation		3			3	0
	2-4-7-14-15-23 1 operation each	1	4			7	6
21	2 operations		5			4	0
17	10 operations	8	1	10	4	49	4
6	3 operations		4			3	3
9	2 operations		1				6
3	5 operations		5			1	5
8	4 operations		4			1	3
10	1 operation		1				3
12	2 operations		4	1	4	1	0
	Total	25	2	25	0	159	2

TABLE NO. 2.

Labor in Assembling.

Part No.	Operation.	Toolmaker. Days.	Hrs.	Journeyman. Days.	Hrs.	Boy. Days.	Hrs.
1 and 2	Putting in pins		1				6
	Filing off pins			1	0		
			1	1	0		6
5-6-7-8-9-10 and 11	Forcing internal gears in place		2	2	0		
	Putting pins 7 into 6					2	0
	Putting together all parts and screwing on 10		1	1	0	1	0
			3	3	0	3	0
	13-14 Forcing in pin 14		1			1	0
	13-15-21 Putting in adjusting screw 21 and retaining pin 15		2	2	0	2	0
	13 and 20 Putting in part 20 and screwing home		2	1	0	1	0
	17-18 and 19 Forcing in bushings		1	1	0		
	13-16 and 17 Putting together		2	1	0		
		1	0	5	0	4	0
3 and 4	Riveting knob to front		1			1	0
	Riveting front and box		3			4	0
			4			5	0
All parts.	General assembling, testing and adjusting	1	0	10	0	10	0

SUMMARY.

Group	1		1	1	0		6
	2		3	3	0	3	00
	3	1	0	5	0	4	00
	4		4			5	00
General		1	0	10		10	00
		3	0	19		22	6

Taking now the various items of cost we have:

Labor for machining 25 days 2 hours toolmaker at $3.50	$88.38
Labor for machining 25 days journeyman at $2.50	62.50
Labor for machining 159 days two hours boy at $1.25	199.07
	$349.95
Labor for assembling—3 days toolmaker at $3.50	$10.50
Labor for assembling—19 days journeyman at $2.50	47.50
Labor for assembling—22 days 6 hours boy at $1.25	28.43
	86.43
	$436.38
To this we will add to cover shop superintendence and possible errors in estimating 15 per cent	65.46
	$501.84
Cost of materials and work done outside	$480.00
To which we will add a profit of 10 per cent	48.00
	528.00
	$1,029.84
On the item of labor cost we will add a profit of 33 1-3 per cent.	167.28
	$1,197.12

This makes the price for a single machine a trifle less than $1.20, but we will put in our bid at $1.25 each.

The cost of tools would now be taken up and estimated in detail in much the same manner, but I shall give here only a list with the prices that we will charge the customer.

LIST OF TOOLS.

Part 1.	Four metal patterns, gated	60.00	
	Metal core box	25.00	
	1 pair chuck jaws	5.00	
	Boring tool—roughing	20.00	
	Boring tool—finishing	20.00	
	Counterbore	2.50	
	Drill jig	15.00	
	Gages	20.00	$167.50
Part 5.	Eight metal patterns, gated	25.00	
	1 pair chuck jaws	5.00	
	Boring tool—roughing	25.00	
	Boring tool—finishing	25.00	
	2 sets milling jaws	10.00	
	Interlocking milling cutters (2 pairs)	15.00	
	Drill jig	18.00	
	Tapping fixture	3.00	
	Special machine for bayonet slots	75.00	
	Gages	25.00	226.00
Part 13.	Eight metal patterns, gated	40.00	
	1 pair chuck jaws	7.00	
	Mandrel	2.00	
	1 pair chuck jaws	7.00	
	Counterbore	2.00	
	Milling jaws	5.00	
	Cutters and arbors	15.00	
	Drill jig	40.00	
	Reaming jig	15.00	
	Gages	20.00	153.00
Part 22.	12 metal patterns, gated	25.00	
	2 drill jigs	15.00	
	Counterbore	2.00	42.00
Part 16.	Screw machine box and forming tools	25.00	
	Precision lathe special rest	15.00	
	Milling fixture for cutting teeth	20.00	
	Gages	15.00	75.00
Part 20.	Forming tools	10.00	
	Reamers	7.00	
	Arbor for turning	3.00	
	2 sets milling jaws	6.00	
	Cutter and arbor	10.00	36.00
Part 18.	Forming tools		5.00
Part 19.	Forming tools		5.00
Part 4.	Forming tools		10.00
Part 21.	Box and forming tools		15.00
Part 17.	Milling fixture	20.00	
	Cutter and arbor	12.00	
	Hollow mill	3.00	
	Grinding fixture	40.00	65.00
Part 6.	Shearing die	8.00	
	Milling jaws	12.00	
	Cutter and arbor	8.00	
	Drill jig	8.00	36.00
Part 9.	Cutting shear		6.00
Part 3.	Blanking and piercing die	20.00	
	Stamping punch	15.00	
	Blanking die	20.00	
	Forming die	30.00	
	Forming die	20.00	105.00
Part 8.	Blanking die	15.00	
	Two piercing dies	25.00	
	Forming dies	8.00	48.00
Part 10.	Blanking and piercing		25.00
Part 12.	Blanking die	30.00	
	Die for cutting teeth	75.00	105.00
	Total		$1,124.50
	In addition to the above there are needed several fixtures for use in assembling and we will add to cover these		75.50
	Making the total for the tools		$1,200.00

The writer thinks the above covers the example selected quite fully and submits it in the hope that it may be thoroughly studied by many of MACHINERY'S young readers.

* * *

Some definitions of early writers are entertaining, if not useful. One such is: "Iron is made up of salts of vitriol and earth badly combined"; another is: "Steel is of two forms—natural and made. When well cleaned and quenched several times in an extract of water and radishes, which has contained earth worms, it cuts iron like lead."

SPHERICAL BORING AND TURNING IN THE ALLIS-CHALMERS SHOPS.

The building of huge stationary engines with shafts of large diameter carrying flywheels and other rotating elements of great weight between bearings widely separated, has introduced the spherical or self-aligning bearing into large stationary engine practice generally. A nickel-steel shaft carrying a load, say, of 100 tons, 10 feet or more between bearings,

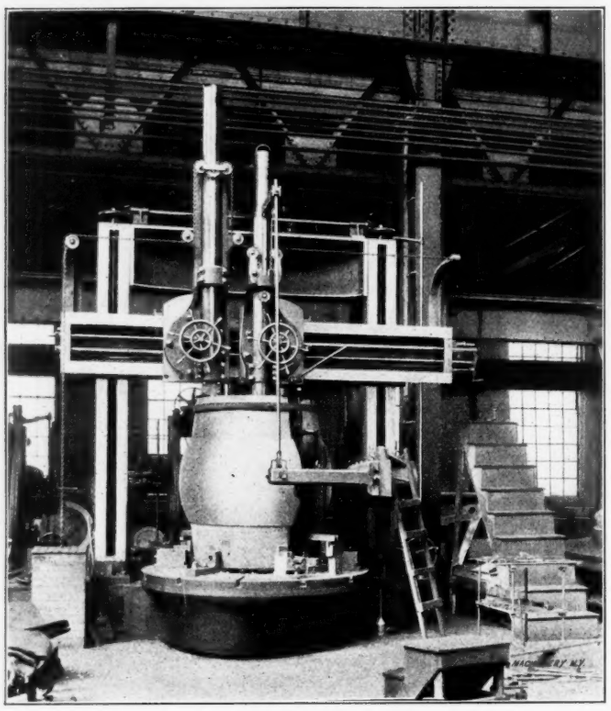


Fig. 1. Boring Mill with Rig for Turning Spherical Shells.

must be as elastic as a finely tempered steel spring, since its deflection in the center under the enormous load causes a constant bending action which may be repeated millions of times in its lifetime. It would be difficult to predict exactly what this deflection will be under all the varying conditions of load, etc., and if it were possible to do so it would be impracticable to bore the bearings at the proper angle to take care of the shaft deflection. Moreover, with bearings four feet long or more, a small variation from the calculated deflection would throw an enormous unit load on the sides of the bearings. So

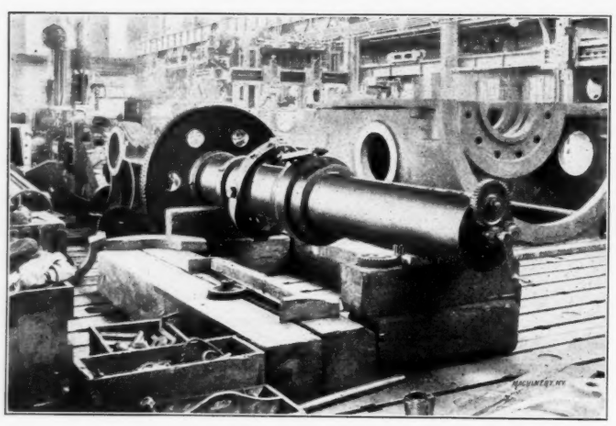


Fig. 2. Boring Bar for Boring Spherical Pillow Blocks, showing Feed Motion.

the deflection must be taken care of in some way automatically; this is done, as stated, by boring the pillow-blocks spherical and making spherical shell bearings for the shaft.

In the Allis-Chalmers shops at West Allis, an 8-foot Niles boring mill has been fitted up with an interesting rig for turning spherical shells, which is shown quite clearly in Figs. 1 and 4, herewith. A vertical slide is fitted to the stanchion or housing at the right, which carries another slide having a horizontal movement at right angles to the cross-rail. This slide is pivoted like a lathe compound rest, so that it may be swung vertically in the arc of a circle, and it carries a curved arm which projects over to the central plane of the mill. The pivot of the swinging slide is so located that the prolongation of its axis intersects the vertical axis of the boring mill table. At the outer end of the arm is the turning tool, which, as it is swung through the arc, generates a sphere on the surface being turned. The motion of the arm is controlled by the boring bar at the right, a rod being connected to the tool-post end and to the upper end of the boring bar, as indicated clearly in Fig. 4. At the upper end of the boring bar a double

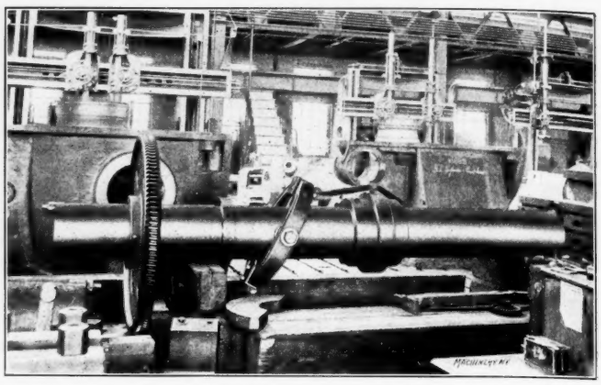


Fig. 3. Boring Bar for Spherical Boring, showing Driving Gear.

mortise is cut, in which a horizontal bar is fitted; and to this is connected the vertical rod which in turn is attached to the end of the swinging arm. The connections are pivoted so that the slide, on which the arm is mounted, may be moved in or out to accommodate shells of different diameters. The feed, of course, is taken from the boring bar.

The complementary tool, the boring bar for boring the pil-

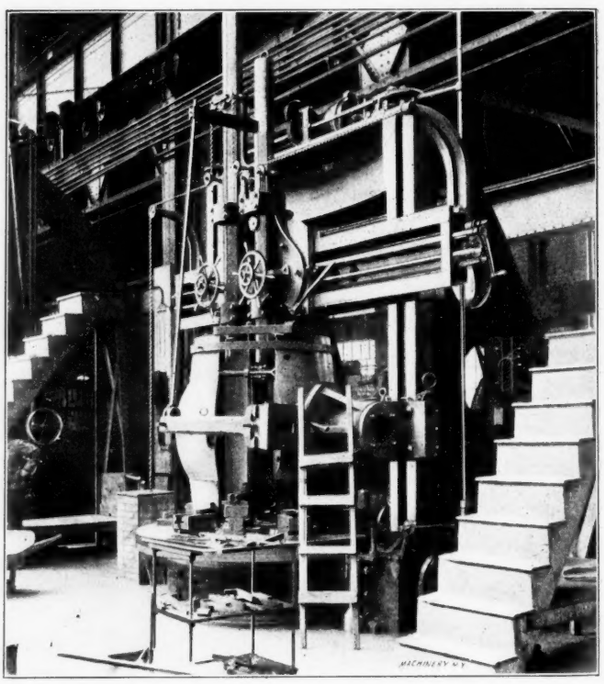


Fig. 4. View of Rig, showing Method of giving Tool Vertical Circular Feed.

low-blocks to the corresponding spherical shape, is shown in Figs. 2 and 3. The bar is driven by a worm and wormwheel, and the boring tools are carried on opposite ends of a loop-shaped swinging arm which is pivoted to the bar in its center. The swinging arm is connected by a link to the sleeve at the right, which is engaged by means of a nut with a feed screw lying in a longitudinal groove in the bar. The gearing oper-

ating the feed motion is clearly shown in Fig. 2; the feed motion is made automatic by simply holding the central pinion stationary while the bar revolves. The rig is substantially the same as that illustrated in MACHINERY in February, 1905, and which we believe was used in the Dickson works of the Allis-Chalmers Company at Scranton, Pa., for boring the pillow-blocks for the Brooklyn Rapid Transit Company's engines; but, in this case, the swinging arm is provided with tool posts at both ends so that cutters may be used on opposite sides of the bar to balance the thrust of the cut. The 12,000 H. P. engines for the Manhattan Elevated Railway and for the New York Subway power stations are fitted with spherical bearings which, we understand, were turned and bored with the tools illustrated.

* * *

TOOL CARRIERS.

OSCAR E. PERRIGO.

The speedy and economical transportation of small tools to and from the general tool room is a problem that, while of acknowledged importance in any large establishment, is of sufficient moment in many shops of more moderate dimensions, and should receive a larger share of consideration than is generally given to it. The requirements of modern shop practice that the employees should not leave their machines to grind tools, or to go to the tool room to procure or to return them, becomes a large factor in the problem of their quick transportation, and even with the system of errand boys to do this work there are many conditions of the modern shop where a good and efficient system of transportation in both vertical and horizontal directions becomes a necessity, if we consider economy along with efficiency. For instance: In a large plant of one floor, it may not be convenient to locate the tool room near the center of the machine shop for the purpose of shortening the lines of travel of the errand boys. Or, if so located, these lines may still be so long as to cause a considerable loss of time. In this case the establishing of auxiliary tool rooms, while of considerable advantage in the distribution of tools, still leaves the transportation problem untouched and out of the question.

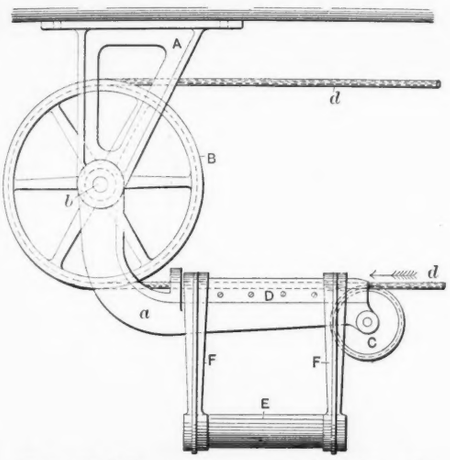
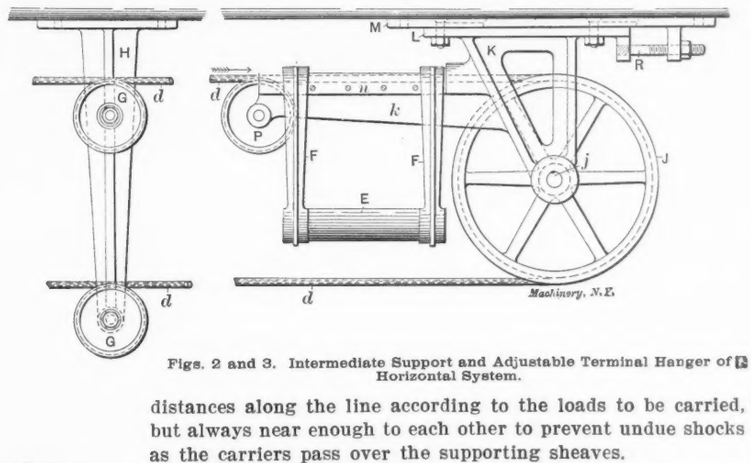


Fig. 1. Fixed Terminal Hanger of Horizontal Tool Carrying System.

Again, in shops consisting of several floors, as many in the crowded cities must necessarily be, the vertical transportation of tools from a general tool room to and from the several floors should be accomplished as quickly and with as little manual labor as may be.

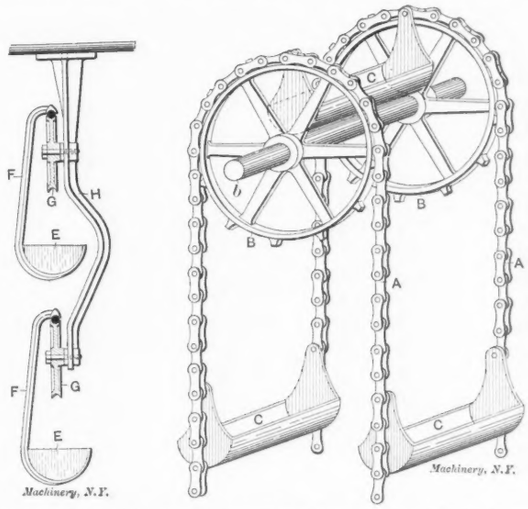
These being the conditions, it becomes necessary to devise a system of transportation that will accomplish the required results in as economical and efficient a manner as possible, and at the same time occupy as little space in the shop as may be, and that shall not be liable to frequent interruptions from getting out of order. It must, therefore, be some overhead system, when used for horizontal transportation; it must be simple and of as few parts as possible so as to be inexpensive, and to be less liable to disarrangement; and it must practically take care of itself under all ordinary circumstances.

It is believed that the system herein shown and described, if properly constructed and installed, will fulfill all these conditions satisfactorily. In the engravings accompanying this article, Figs. 1, 2, 3, and 4, represent the system of horizontal transportation, and Fig. 5, that for vertical service. The horizontal system consists of a half-inch braided cotton sash cord, fitted with metal couplings and passing over comparatively large sheaves at each terminal. Upon this carrying cord are suspended hanging receptacles of suitable form and dimensions for holding the tools to be transported. Fig. 1 is a side elevation of the fixed terminal, the shaft *b*, upon which is fixed the main sheave *B*, being journaled in the hangers *A*, one of which is of sufficient length to furnish a bearing for the shaft, while the other has formed upon it a curved arm *a*, in the outer end of which is journaled the sheave *C*, for the purpose of supporting the carrying cord *d*, at the point where the carrier leaves the cord in coming to rest. Fixed to the extending arm *a* is an inverted U-shaped piece of sheet metal *D*, covering the carrying cord *d*, and furnishing a resting place for the carrier. This carrier is composed of a box *E*, of sheet or cast metal, with a curved bottom as shown, and suspended by the malleable iron supports *F F*. These are shown in end elevation, the supporting sheaves in section, in Fig. 4, which shows the supporting sheaves *G G* and the bracket *H*, on which they are journaled. This device is also shown in front elevation in Fig. 2. This supporting device should be located at suitable distances along the line according to the loads to be carried, but always near enough to each other to prevent undue shocks as the carriers pass over the supporting sheaves.



Figs. 2 and 3. Intermediate Support and Adjustable Terminal Hanger of Horizontal System.

The carriers shown are of simple form and will be most useful for carrying the usual variety of small tools. Any convenient form may be used, however, the center of gravity being always kept directly under the carrying cord *d*. They may be made with considerably less drop from the carrying cord, for most kinds of tools, this distance being reduced nearly one half, which will cause them to move with less shock and less of the swinging motion as they pass along. They should never be relatively lower from the carrying cord than is here shown.



Figs. 4. and 5. End View of Intermediate Support for Horizontal Transportation, and Carriers, Vertical System.

Fig. 3 shows the adjustable terminal hanger *K*, and its connections. In this case two hangers are cast on a plate *L*, or permanently fixed to it, one of them having a projecting arm *k*, carrying the supporting sheave *P*, and being provided with the U-shaped cord shield *n*, similar to that shown at *D*, Fig. 1. The plate *L* is arranged to slide upon the plate *M*, and is secured to it by two bolts as shown. The adjusting screw *R* is provided for taking up any slack that may be in the carrying cord *d*. The main sheave, *J*, is fixed upon the shaft *j*, which is journaled in the hangers *K*.

In operation, the device is driven from a pulley of suitable size on the shaft *b*, and at such a speed as will give sufficient momentum to the carrier *E*, to cause the carrier arms *F F* to ride up on the whole length of the cord shields *D* and *n*, and stop there with very little shock, the carrying cord running in the direction of the arrows. In use the carriers have only to be taken off and hung upon the returning portion of the carrying cord to return the empty or loaded carrier. Upon the arrival of a carrier it glides up on the cord shield *D* or *n*, and remains there until removed. Should a second carrier arrive before this one is removed no harm is done, although it is expected that they will be removed as soon as they arrive. The cord shields *D n* may be made long enough to accommodate two carriers, but this will seldom be found necessary.

The vertical system is shown in Fig. 5 and consists of two parallel chains *A A*, passing over the sprocket wheels *B B* at the top and under similar ones at the bottom. These chains may be driven from either the upper or lower shaft as may be most convenient. Pivoted to these chains are the carriers *C C C*, which may be made of cast or sheet metal, and so formed, with the center of gravity considerably below the pivots, that they may always remain right side up, even when passing over the shaft *b*. These carriers should be about eight feet apart on the chains, and they should be painted a different color for each floor to which they are consigned, so that their contents may be readily removed at the proper destination without stopping the movement of the device for that purpose, it being understood that all descending carriers are consigned to the tool room on the first floor, or toward it, provided it is located on any other floor. These carriers may be so constructed as to automatically dump their contents at the proper floor without the attention of an attendant for that purpose. This system may be used in situations where continued vertical and horizontal transportation is wanted, by running the chains over gliding sprocket wheels at the proper turning points, as the carriers will always maintain their proper positions no matter what may be the direction of movement of the chains. However, for long horizontal distances this system will not be found as economical or as efficient as the first method described. The chain system should be run at a much slower speed than the cord system, as the carriers should be unloaded while in motion, while those on the cord system come to full stop until they are again wanted. Each will be found to be best adapted to its own particular sphere of usefulness as herein described.

AN ENGLISH HIGH-SPEED PLANER.

Our British friends seem to have been making considerable progress along lines of their own in the problem of designing high speed planers. The Bateman's Machine Tool Co., Ltd., of Leeds, England, have gotten out a line of patented high-speed planers, which carries several points of considerable interest. One of these machines, the 60-inch by 60-inch by 12-foot planer, here illustrated as Fig. 1, has a cutting speed of 40 to 45 feet per minute, and a return speed at 140 to 150 feet per minute, certainly a remarkable speed for a planer of its size. The rate of speed is correspondingly high on the smaller planers. On the 24-inch by 24-inch by 6-foot machines, the cutting speed can be arranged from 20 to 60 feet per minute or higher if required, with the return speed of 170 to 220 feet per minute. These high speeds are said to be characterized by very smooth running and the absence of any very great comparative absorption of power at the moment of reversal. These good points are largely produced by the patented features of the machine, the sliding rack and the flywheel loose pulleys for use with the overlapping belt, both shortly to be described.

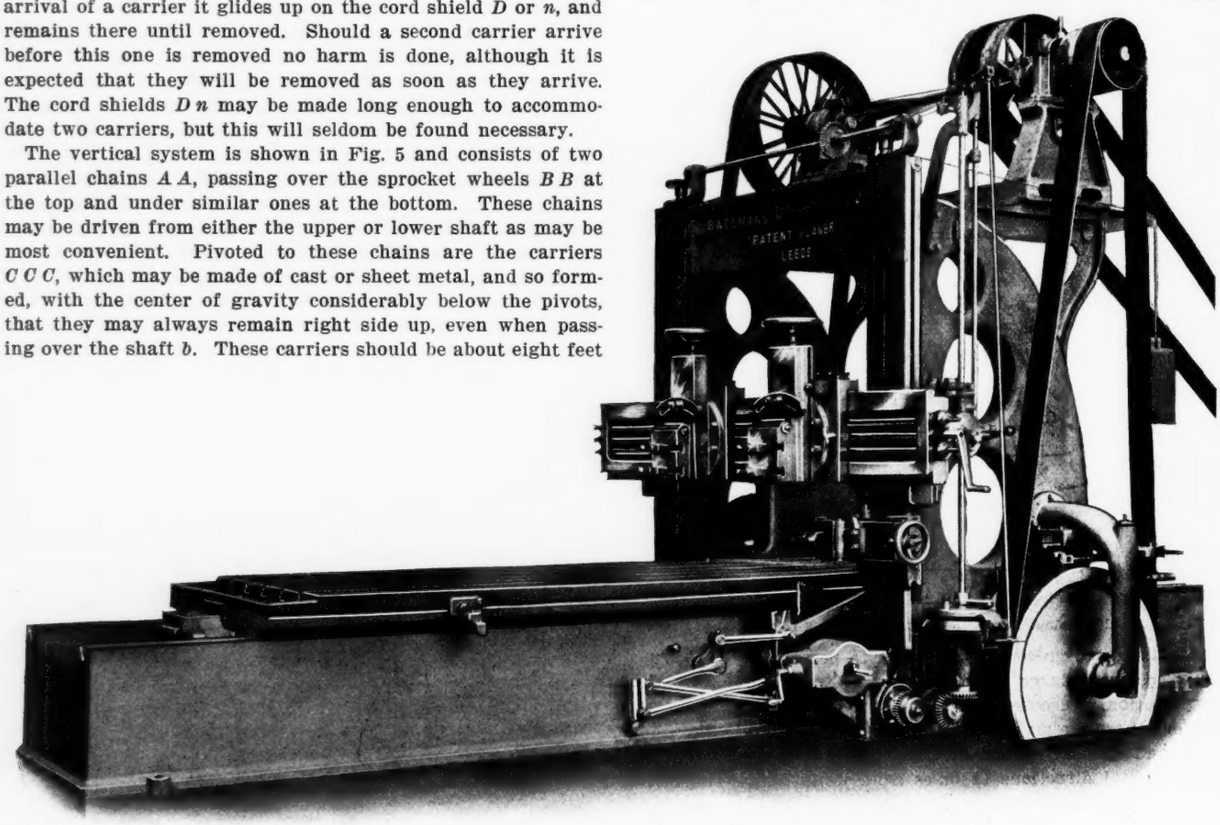


Fig. 1. Bateman's 60-inch x 60-inch x 12-foot Planer, Cutting at 45 feet per minute, and Returning at 150 feet per minute.

In the ordinary rack-driven planing machine, the rack is bolted direct to the table. In Bateman's planer a patented feature in the form of a sliding bar which has a short longitudinal motion in a groove in the under side of the table is introduced, and the rack is fixed to this bar. This arrangement is shown in Fig. 2, which gives a plan of the under side of the planer table. To a crosshead at one end of the sliding bar, spindles are secured, carrying adjustable springs. These springs absorb the shock of reversal and, being compressed, help the table to start in the new direction at reversal, relieving the belts of much of the extra strain to which they would otherwise be subjected. There are two sets of springs fitted underneath the table, one to absorb the reversal from return to cut and the other pair to absorb it from cut to return. If the springs are properly adjusted the table floats between them, and as soon as the cut is started one pair of springs tends to be compressed. The adjustment of this pair of springs can be so arranged that the pressure of the tool on the work brings the crosshead at the end of the rack in

direct connection with the end of the table. There is thus a metal to metal pressure, avoiding the possibility of any unevenness in the cut. The makers say, however, that they find it unnecessary to make this adjustment in many cases, especially if the cut is light.

The second patented feature making for smoothness in running is the use of flywheel loose pulleys, used with overlapping belt and automatic friction clutches between the flywheels and fast pulleys, there being one clutch on either side of the machine for cut and return. The operation of this device may be understood by the sectional view given in Fig. 3. The belts, being wider than the fast pulleys, overlap onto the flywheel pulleys when thrown over to do their share of the work, and through this overlap obtain much of the conserved energy of the flywheels at the moment when it is most required. The friction clutches further assist the fast pulleys in obtaining conserved energy from the flywheel loose pulleys.

The action of the clutch is briefly as follows: The rod *A* is actuated by the belt shifting devices and moves the vertical shaft *B*, which carries at its lower end a small pinion *D*.

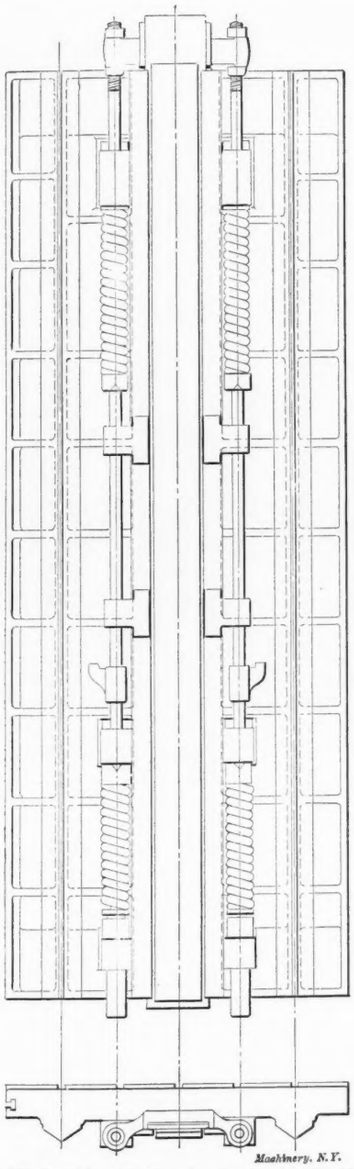


Fig. 2. Plan of Under-side of Table, showing Device for Absorbing Shock.

This latter gears into a short rack cut on the slidable bushing *E*, the bushing being free to move along the pulley shaft *F*. Attached to the inner edge of the slide bush *E* are segments of the gripping ring *G*, working in a recess on the boss of the flywheel loose pulley *H*. A ball thrust bearing is provided between the end of the slide bush and of the boss of the flywheel pulley. This loose pulley *H*, as well as the slide bush, is free to move along its shaft. Any motion given to rod *A* by the shifting devices of the planer is, as is easily to be seen, transmitted to the flywheel, which is thereby moved so that the friction clutch *J* is either brought into or brought out of action. This motion is so timed as to take place when the belt is thrown on or off the fast pulley, and, together with an overlapping of the belt from the fixed on to the flywheel pulley, enables the conserved energy of the flywheel to be transmitted into the fixed pulley at the moment when most power is required as stated above.

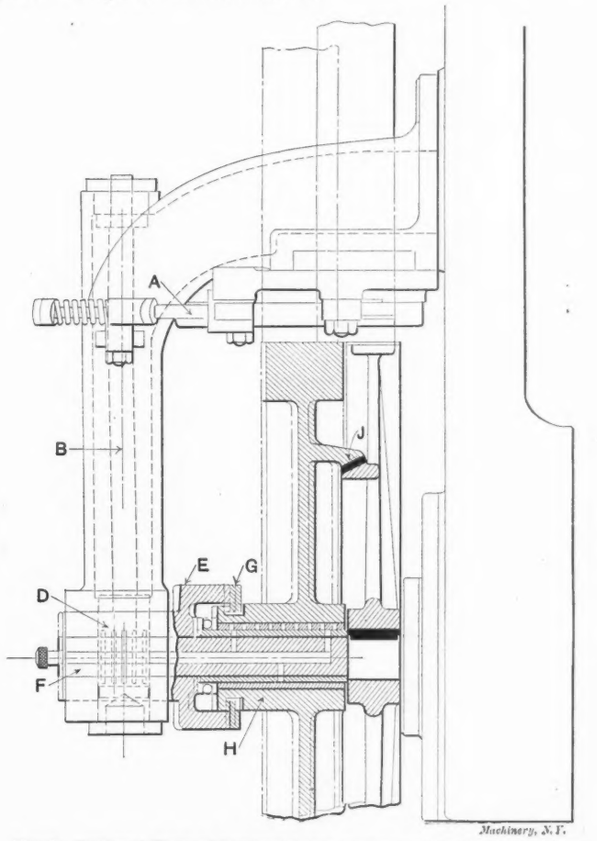


Fig. 3. Sectional View of Flywheel Loose Pulley, used with Overlapping Belt, and Automatic Friction Clutch.

The beds of these planers are heavily designed and are of double box form without feet. They are unusually long, the run-over of the table consequently being very short. The tables are strongly webbed to counteract any tendency to buckle or distort. The housings are designed for great rigidity and are tied together at the top by a bridge piece. The cross rails are deep and can be raised and lowered by power as well as by hand on all except the smaller machines. The length of cross rail provides that where two tool heads are used, the near head can be traversed across the whole width between the housings independently of the off head. All planers of medium or large size are built to carry two tool heads on the cross rail and one side tool on the near housing, and can be fitted with a side tool head on the off side if desired. The feeds for the cross rail tool heads are automatic and can be started, changed or stopped instantly while the machine is at work, and are actuated by a new friction arrangement through a vertical shaft carried on ball bearings. The vertical feeds on the side tool heads are actuated by the knocking-off arrangement. The high speed gearing is chiefly of cast steel and is lubricated in an oil bath; the other gearing is of cast iron, and all are specially broad on the tooth. The high speed shafts are of carbon steel and run in phosphor bronze bearings.

* * *

It is alleged that Dr. Hadfield has discovered an alloy of aluminum, copper and manganese which is nearly as magnetic as low-grade cast iron, although it contains no iron at all. The discovery is considered as being of importance, as it may throw considerable light upon the mystery of magnetism.

AN AUTOMATIC GEAR CUTTING MACHINE.

An automatic and universal gear cutting machine, having a number of new features, has been placed on the market by the Brown & Sharpe Manufacturing Co., Providence, R. I. The machine will cut spur and bevel gears of 12 inches diameter, 8 inches face, and 10 diametral pitch, if of cast iron, or 12 diametral pitch if of steel.

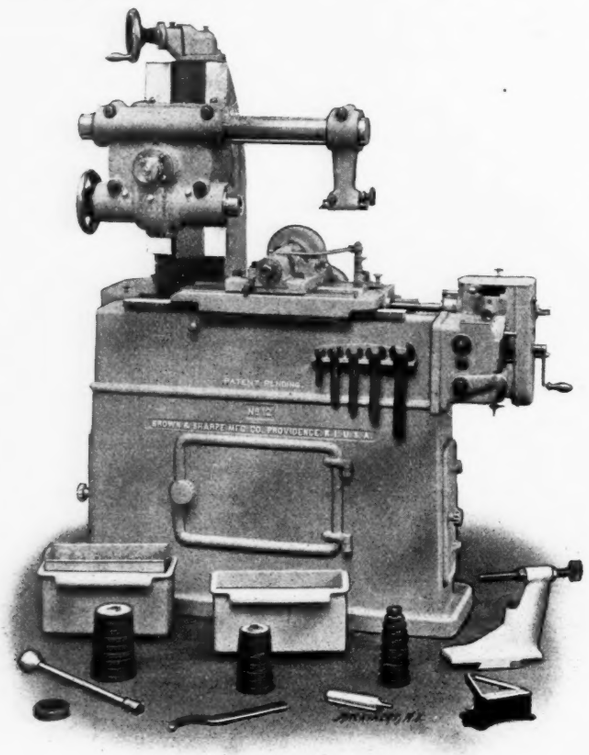


Fig. 1. Front View, Automatic Gear Cutting Machine.

Reference to the front and rear views, Figs. 1 and 2, shows the frame of the machine to resemble the type that has already been used by this company, but close inspection will reveal a number of important differences in the design of the working parts. First of all, the work spindle and the outer support for the work are carried by a swivel head, graduated to allow the spindle to be set to any angle or to be reversed in position, so that the work may be placed on either end of the spindle. The index wheel, apparently conspicuous for its absence, is encased within the head where it is protected from dirt and chips. In Fig. 2 it will be noticed that the drive for the cutter spindle is by means of sprockets and a silent

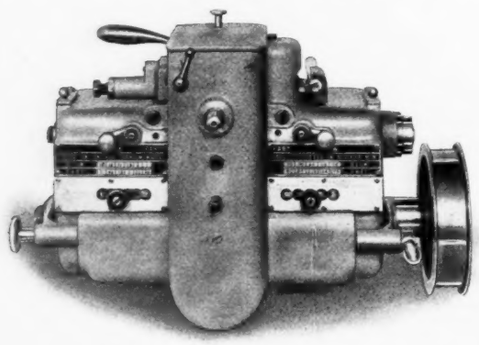


Fig. 3. Speed and Feed Gear Case.

chain. At the end of the machine is a large gear box, on one side of which is a single driving pulley which operates all of the different mechanisms of the machine. This pulley runs at a constant speed of 300 revolutions per minute and is driven by a 1¼-inch belt. The gear box contains two sets of cone gears for the feeds and speeds. Motion is also transmitted from this gear box to the quick-return mechanism and the indexing mechanism, which controls the working spindle.

The line engraving (Fig. 5) shows the section of the head which carries the work spindle and the upper part of the upright, upon which the head slides. Motion is transmitted to the work spindle from the shafts *M* and *N* and the dividing wheel *W*, in which a worm upon shaft *N* meshes. The head is adjustable on the machine upright and is controlled by means of an elevating screw which is operated by a hand wheel, the thrust being taken by ball bearings. A dial graduated to read to thousandths of an inch indicates the position of the head and facilitates setting the blank for depth of cut.

The head can be completely revolved about the axis of shaft *N* to allow the other end of the work spindle to be used, which adapts the machine to cutting all varieties of bevel gears. The head is graduated around its entire circumference to read

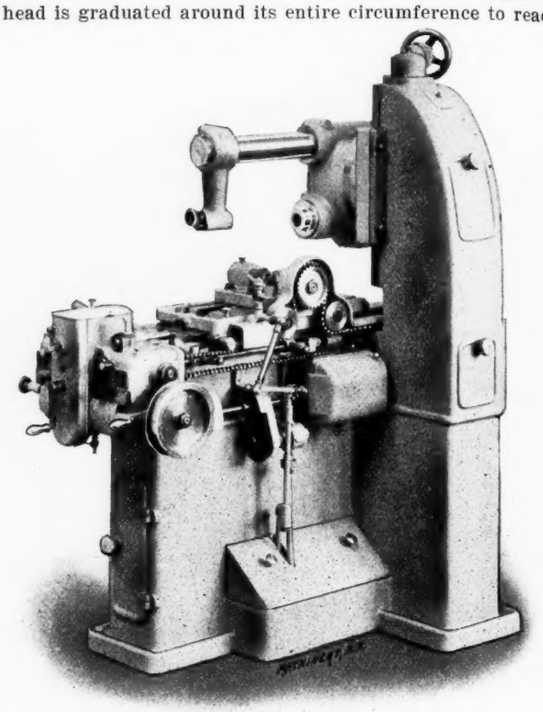


Fig. 2. Rear View of Machine.

to one-half degrees. Both ends of the spindle are provided with a No. 10 taper hole, and the advantage of being able to use either end of the spindle will be evident from Fig. 8, which shows a method of cutting a bevel gear on the end of a long arbor. The worm and wheel are not disengaged when adjusting the work; the worm is held to its bearing on the shaft by a friction cone and the locking screws *A* and *B*

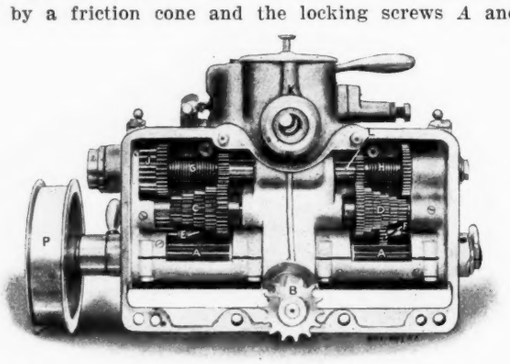


Fig. 4. Rear View of Case, showing Gears.

(Fig. 5) which, when loosened, permit the worm to be turned by the knurled knob, *C*. The base of this knob is graduated to read to thousandths. The hand wheel on the end of the work spindle serves to release a friction clutch and allow the spindle to be turned by hand when truing up the work. An overhanging arm, *K*, Fig. 5, is clamped in the head and supports the outer end of the work spindle; while, in addition

to this, an adjustable rest is provided which can be fastened to the frame of the machine directly back of the rim of the gear in front of the cutter, and serves as a support when cutting spur gears.

In Figs. 3 and 4 are exterior and interior views of the case containing gearing to control the speed and feed of the cutter.

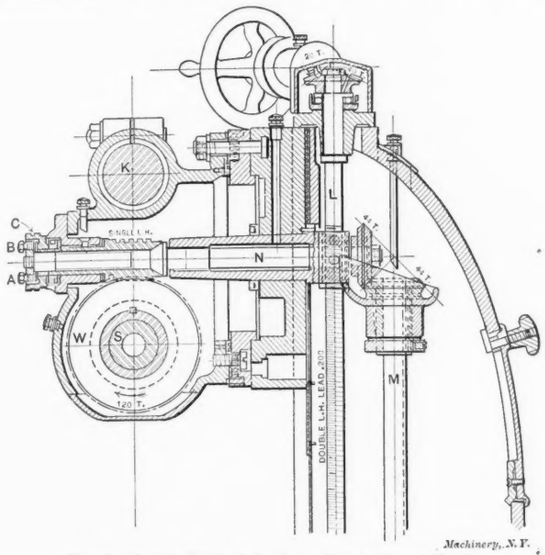


Fig. 5. Partial Vertical Section of Upright and Head.

The drive is through the pulley *P*, Fig. 4, and the shaft carrying the elongated pinions *A A*. The one of these at the left transmits motion to the sprocket wheel *J*, which drives the cutter by means of a silent chain; and the other to the shaft *L*, which drives the feed mechanism contained in the upper

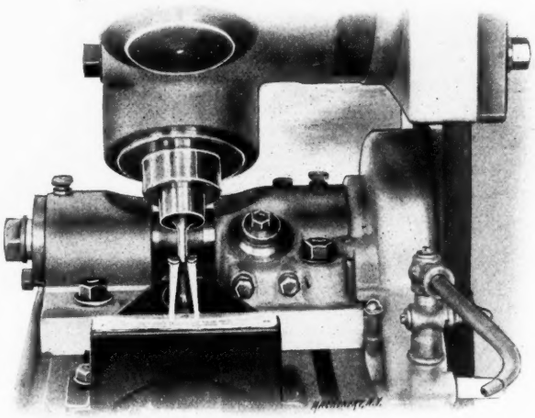


Fig. 6. Cutter Indicator in Position.

part of the casing at *K*. The speed and feed mechanisms, therefore, are entirely separate, and are operated from a constant speed shaft; and when the speed of the cutter is changed, the feed is in no way affected unless a change in the feed gearing is also made. The speed mechanism is a cone of five

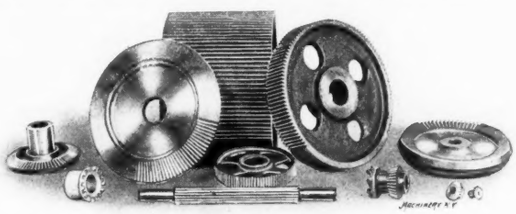


Fig. 7. Some of the Product, showing Capacity of Machine.

gears, *C*, driven through the pinion *A* from the sliding gear *E*. There is also a pair of back gears, doubling the number of speeds. These are attached to the sleeve *G*, which has annular grooves turned on it so that it may be moved longitudinally on its shaft by engagement with a small pinion operated by a handle in front of the casing, for the purpose of bringing either of the gears into mesh with the corresponding gears of the cone. The sliding gear *E* is operated by one of the knobs (also in front of the casing), together with a swinging handle shown at the right, near the pulley in Fig. 3. This combination produces 10 speeds, and the similar arrangement of feed gears produces 12 feeds, varying from ¾ inch to 5

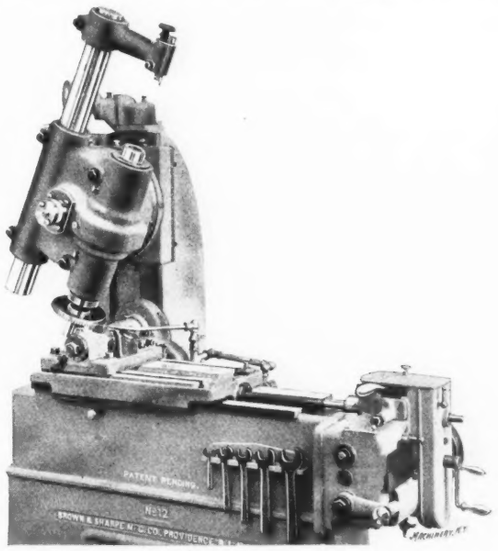


Fig. 8. Cutting Bevel Gear with Inside Hub by using Outer End of Work Spindle.

inches per minute. The engraved plates on front of the gear casing show the speed and feed secured with any given set of mechanisms.

The quick-return motion is driven through a worm and worm wheel by the pulley shaft and a sprocket wheel on the same shaft with wheel *B*, Fig. 4, but at the other end of the shaft. A silent chain connects this sprocket wheel with an-

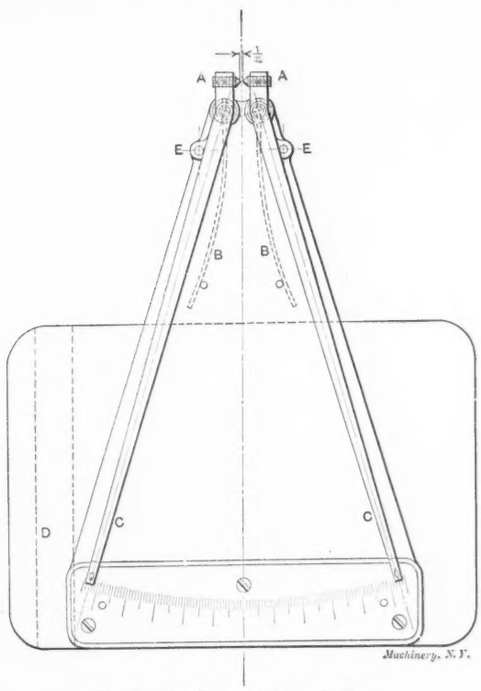


Fig. 9. Plan View of Cutter Indicator.

other sprocket on the feed screw which actuates the cutter slide. The quick-return thus operates at a constant speed without regard to the feed of the cutter.

Sprocket wheel *B* is connected by a chain with the horizontal shaft seen at the rear of the machine in Fig. 2. This shaft operates the spacing mechanism for the work spindle, this mechanism being encased in the cast iron box appearing by the upright and just below the cutter slide in Fig. 2.

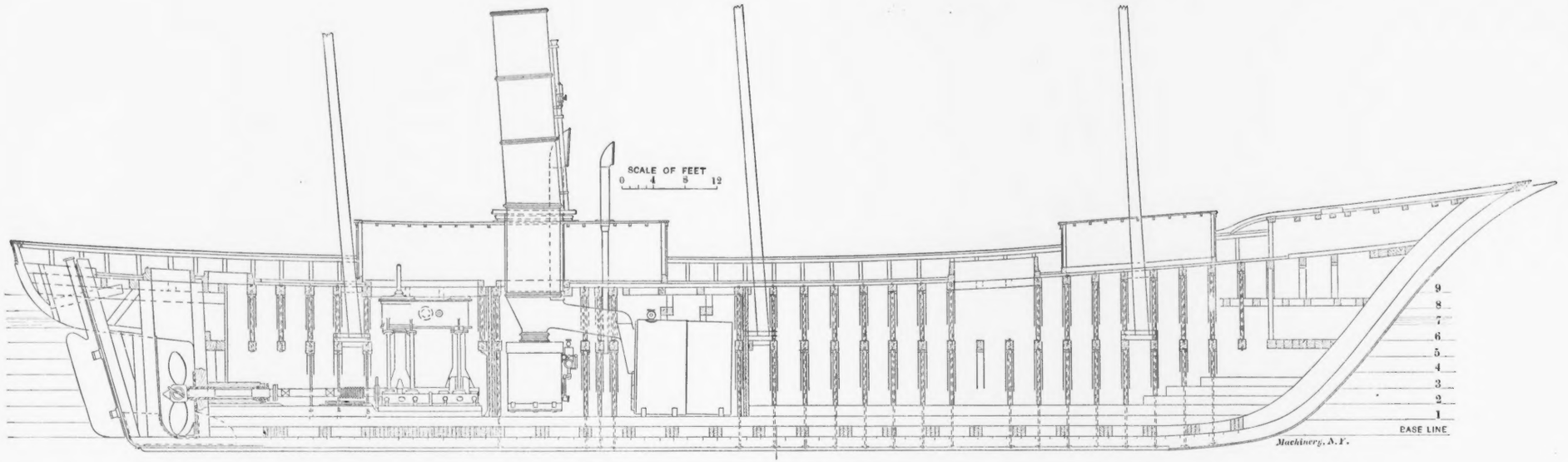


Fig. 2. Longitudinal Section of Hull, showing Heavy Bracing.

The index worm and wheel are entirely enclosed, and the spacing mechanism provides for the cutting of all numbers of spurs or bevels from 12 to 50, and of all numbers from 50 to 400, except prime numbers and multiples. The change gears in the indexing mechanism all have clutches to attach them to their shafts, no keys being used owing to their liability of working loose and creating errors in the indexing.

An indicator for setting the cutter centrally with the work is provided, as shown in position with the test center in the work spindle, in Fig. 6. This indicator, shown in plan view in Fig. 9, consists of two swinging arms and a cast iron base made in box form and sufficiently heavy to hold the indicator in place without clamping. Referring to Fig. 9, the arms *C-C* swing about bearings, *A-A* being the measuring points, which can be adjusted to compensate for wear. The measuring surfaces are held in contact with the cutter, when the latter is being adjusted, by the two flat springs shown at *B*. Stop pins *E-E* prevent the arms from swinging too far apart. The tongue *D*, on the bottom of the base, fits into the outer T slot on the cutter carriage.

* * *

THE MACHINERY AND HULL OF PEARY'S NEW ARCTIC SHIP.

Lieutenant Peary, the famous Arctic explorer, is to start on another expedition this summer in search of the North Pole. He and his party will go in the steamship Roosevelt, a new vessel that has been built in Maine expressly for this voyage. The hull of the vessel is of oak heavily braced by oak and hard pine timbers, and through the courtesy of Mr. H. K. Griggs, Portland, Me., we are able to show a number of views of the hull and machinery. The hull was built by McKey & Dicks, Buckport, Me., and the machinery by The Portland Company's works, Portland, Me.

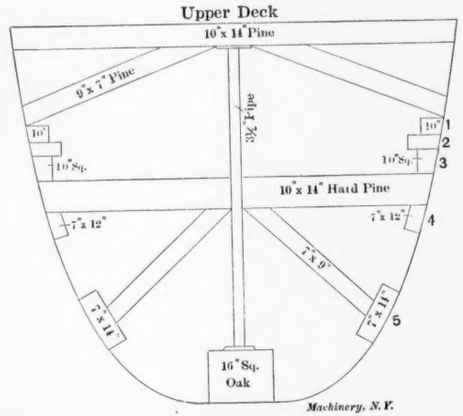


Fig. 1. Section Amidships, showing Dimensions of Braces used in Hull.

The ship is 180 feet long, 32 feet beam, and has three masts, being rigged as a three-masted schooner. She will ordinarily proceed under sail, but the auxiliary steam power may be used as required either to hasten the rate of speed or to force the vessel through the ice floes of the North. Unlike most ships with auxiliary power, however, the engine power is great enough to enable the vessel to proceed under steam at a good rate of speed, and not depend upon her sails, if such a course should be necessary or desirable. A general view of the hull in Fig. 3 was taken before the vessel was launched and shows Lieutenant Peary standing in the foreground. Fig. 2 is a longitudinal section of the hull, indicating the location of the boilers and machinery, and showing the large number of cross and longitudinal braces that are used to strengthen the hull, these braces being additional to the usual framing employed in the hulls of ships of this size.

In the midship section, Fig. 1 is an outline of the arrangement of the bracing with the sizes of the timbers given, thus affording an idea of the enormously heavy construction used for the purpose of resisting the pressure that the hull is likely to be subjected to when in the ice. Cross braces start at the bow and extend aft as far as the boilers, being spaced about 3 feet apart. The keel is of 16-inch square oak and the timber used to brace the bow is of 16 x 24-inch oak. Outside of the

bow, in the vicinity of the water line, there is an extra thickness of 3-inch planking, which is covered with ½-inch boiler plate. Ordinarily little or no bracing is used inside of the hull of a ship except the framing, so that it will be seen to how great an extent the hull of this boat is reinforced.

The main engine is of the usual compound marine type of about 1,400 H. P. The cylinders are 24 and 52 inches in diameter and 30 inches stroke. The shaft is 12 inches in diameter and carries a propeller 8 feet in diameter with four blades. The propeller has a cast steel head with cast steel blades belted to it. Provision is made for removing two of the blades, since two-bladed propellers are considered safer and more effective for ice breaking. Steam is generated in the single Scotch boiler 10 feet in diameter and 12 feet long with three furnaces, and two Almy watertube boilers. The view, Fig. 5 shows the Scotch boiler being placed in position.

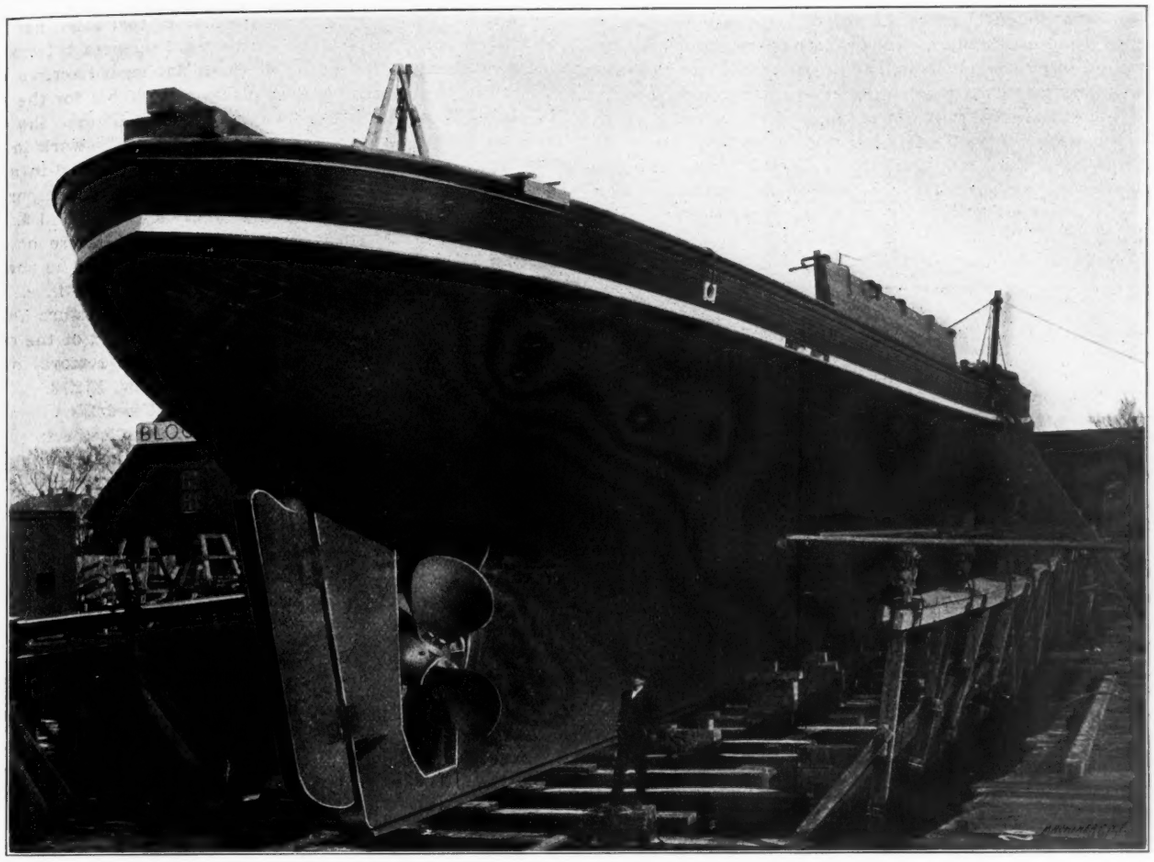


Fig. 3. Hull of S. S. Roosevelt, Peary's New Arctic Ship. Peary Standing in the Foreground. (Copyright, 1905, by W. H. Gay.)

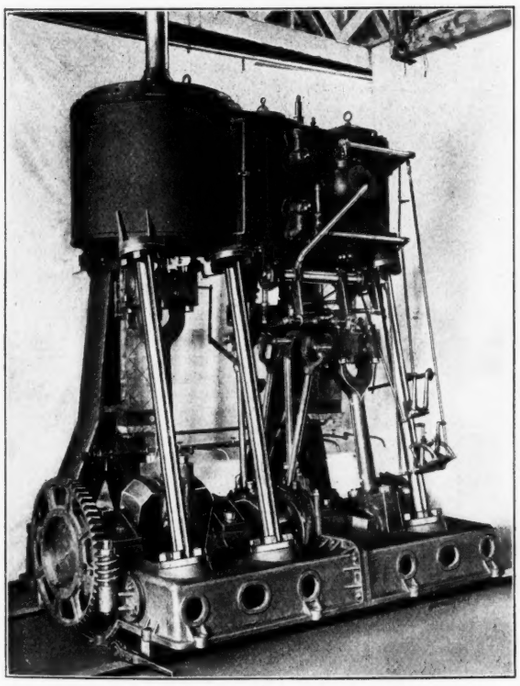


Fig. 4. Engines of 1400 Horse Power used to Propel the Steamship.

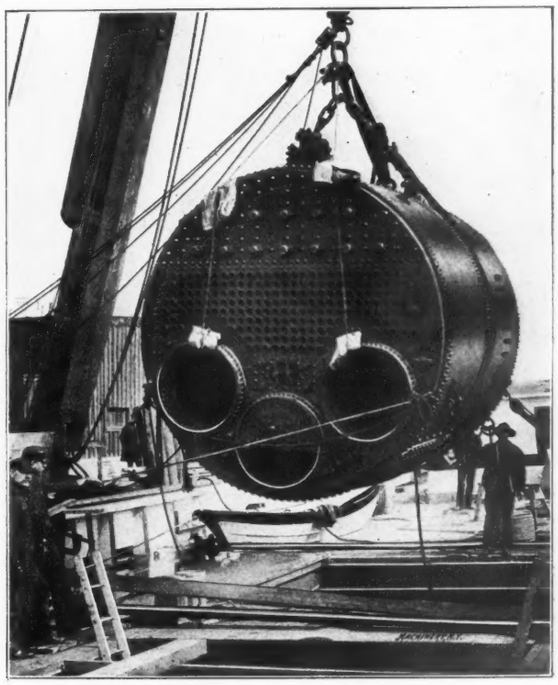


Fig. 5. Lowering one of the Boilers into the Hold.

GANG MILLING OF REPETITION CASTINGS.

JOSEPH V. WOODWORTH.

In the manufacturing of small interchangeable castings for machine parts, gang milling fixtures play a very important part. When the parts are machined to extremely accurate dimensions, and are produced under the modern piece-work system, the object sought is to handle as many castings at a time as possible, in fixtures so designed as to insure the complete interchangeability of the product.

To illustrate the value of gang milling fixtures for manufacturing accurately machined duplicate parts, and also how a number of such parts may be handled and machined expeditiously at the minimum of cost, I describe in this article a gang milling fixture which is in use in an establishment requiring over 100,000 of the castings machined in this fixture per year.

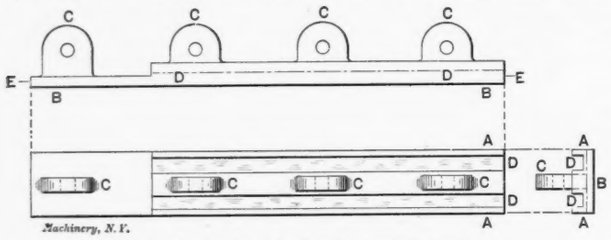
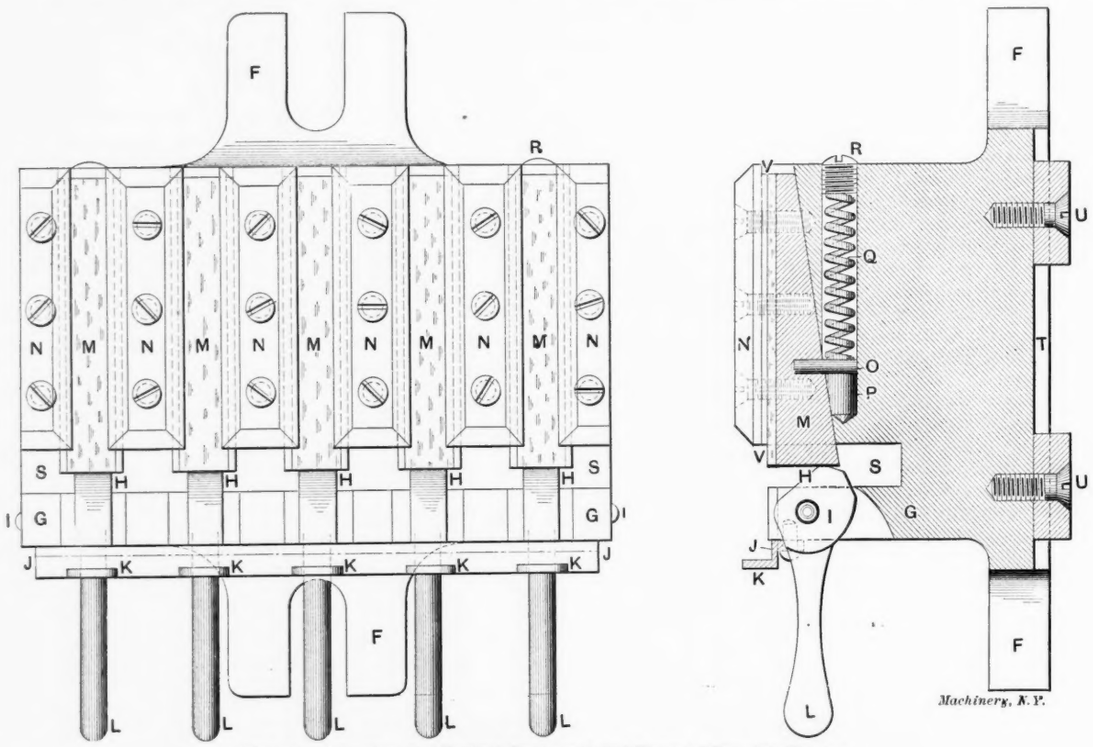


Fig. 1. Casting Machined in Gang Milling Fixture.

In Fig. 1 we have three views of the casting machined in the fixture. The work performed is the milling of the two channels indicated by *D*. Previous to this operation the casting is machined on the back *B* and also on the sides and ends *A* and *E* to limit gage measurements. Subsequent to the operation, the four holes *C* are drilled in the projecting lugs, the insides of the channels being utilized as banking or abutment surfaces for the locating of the castings in the drilling jig. Figs. 2 and 3 are two views of the fixture complete, Fig. 2 being the plan view, which shows the appearance of the fixture without the work in it, and Fig. 3 a vertical cross-sectional view. Fig. 4 is a longitudinal sectional view of the fixture, and also of the gang of ten cutters used in conjunction with it. *Y* represents the cutters; *X* represents the washers or collars; and *W* represents the milling machine spindle. Fig. 5 is an end view illustrating the fixture with the work in position and presented to the cutters for milling.

The fixture handles five castings at a time. The body casting has projections or wings at two sides *F*, and has platen locating tongues at *U*, for fastening and locating it in the platen. The body casting has five inclined channels milled in its face to accommodate the five hardened tool steel work locators *M*. The five parts *N* are also of tool steel, hardened and tempered, and fastened to the wall surfaces between the inclined channels by means of three flat-headed screws each. These pieces serve as banking pieces or surfaces for the work to clamp up against. Five eccentric levers *L* force the work locators up the inclined ways, thus clamping the work in position against the plates *N*. These levers are fastened in milled slots by means of the drill-rod shaft *I*. The eccentric portions of the levers are indicated clearly at *H* in Figs. 3 and 5. *J* is a stop bracket fastened to the back of the fixture or body casting by means of several round head screws. The portions at *K* are stops against which the ends of the castings, to be machined, abut. The construction for forcing the work holders back in the inclined channels upon the releasing of the eccentric clamping levers *L*, thus allowing of the removal of the work, is shown in the vertical sectional view, Fig. 3. It consists of a stiff spiral spring *Q*, located in the drilled hole *P*, a pin *O* for engaging this spring, and the headless setscrew *R*. One end of the spiral spring rests against the screw *R*, and the other against the pin *P*. The tension is kept sufficiently stiff to cause the work-holders to release the work immediately upon the lever *L* being pulled upward; each of the five work holders is equipped with such an arrangement.

When in use the fixture is clamped to the platen of a large universal miller and the table adjusted until the work receivers or holders are in the relative positions to the cutters illustrated in Figs. 4 and 5. The castings are located in the holders; the eccentric levers are pushed downward, as shown in Fig. 5; and the castings are thus clamped in position. The feed is then thrown in and the platen and fixture travel forward until the channels *D* and *E* are milled. The table is then fed backward and the machined work removed.



Figs. 2 and 3. Plan and Vertical Cross-sectional Views of Complete Fixture.

* * *

An English newspaper, the *South Bucks Standard*, tells how a bicycle ball stopped a train: "Recently, as a train was running at high speed on the Great Eastern Railway, the engineer felt that there was something wrong with one of the wheels of the engine. On making an examination he found that a ball of a bicycle bearing had become fixed in the outer surface. The ball was so firmly embedded in the steel that it could not be extracted."

MULTI-STAGE CENTRIFUGAL PUMPS FOR ELEVATOR SERVICE.

An interesting application of multi-stage centrifugal pumps has recently been made in a new and very complete power installation in the building of the John Taylor Dry Goods Co., Kansas City, Mo., and affords another example of the prevailing tendency to use pumps of this type where reciprocating pumps would have been thought necessary only a few years ago.

Three of these pumps were installed for operating the hydraulic elevators of the building under a pressure of 140 pounds per square inch. They are direct-connected to electric motors, receiving their current from generating units comprising part of the power plant of the building. The motors are under the control of Cutler-Hammer automatic controllers mounted on panels and assembled uniformly with the panels of the main switchboard.

The three pumps supply the water for the complete hydraulic system upon which all the hydraulic elevators of the store are connected. The elevators are separated widely over the premises, in some cases nearly a city block apart.

The arrangement of the piping is unique. There is one pressure main from the discharge openings of the pumps, which runs throughout the whole building past each of the elevator machines. This pressure main is connected into a compression tank in each end of the building, the two tanks being set at the same level and carrying their water lines at the same height. The connection for each elevator machine is then taken from this main at a convenient point. There is a discharge or return main paralleling the pressure main into which each elevator machine discharges, and this return main is connected to each of two open, or "surge," tanks located near the compression tanks, and then continues on and connects to the suctions of the pumps. With a suitable arrangement for "snifting" entrained air out of this suction line the operation of this system is perfect.

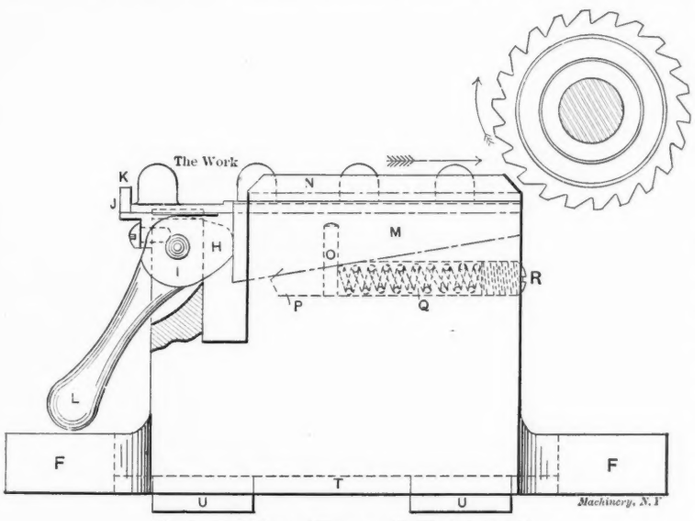


Fig. 5. End View of Fixture with Work in Position.

The capacities of the pumps were chosen to fit the service requirements at all times, throughout the variations of load during a day's run, while at the same time providing a reserve capacity for accidents or contingencies. Two of the pumps have a capacity of 300 gallons a minute and the third one a capacity of 500 gallons a minute. By this arrangement it is possible to secure 300, 500, 600, 800 or 1,100 gallons a minute as required. All that is required for the heaviest service at the present time, however, is 800 gallons per minute, so that with either of the smaller pumps out of service all the elevators can be operated at full capacity, and if the largest pump should break down, 600 gallons a minute would still be available, and would amount to a reduction in capacity of only 25 per cent.

The type of pump was selected from considerations of economy, noiseless operation and great compactness, all of which were very essential in this instance. No reciprocating pump could have been installed in the given space that would have had the requisite capacity with anything approaching the degree of economy guaranteed by the makers of these multi-stage pumps. The importance of compactness will be evident when it is stated that this installation is two floors below the ground floor, in a sub-basement in which there are many building columns and other obstructions. The engineers were not satisfied to install centrifugal pumps, however, until a careful test had been made; and they went to Atlanta, Ga., and tested a similar pump which operates the elevators in the Equitable building of that city, with satisfactory results. The pumps at Kansas City are similar to this one, but have the added feature of the Cutler Hammer controlling apparatus, which automatically starts and stops the pumps to meet the service conditions.

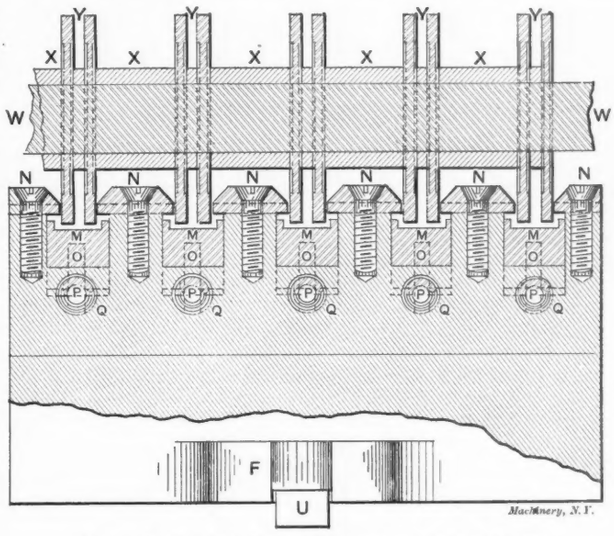


Fig. 4. Longitudinal Section of Fixture and Gang of Cutters.

The nature of the turbine pump is such that a control device is not required except from considerations of economy. The pumps will automatically cease to discharge when the maximum pressure of the system is reached and the current consumption of their motors will fall to that required to merely rotate the mass of water in the impeller chambers. When the pressure is reduced discharge begins and the current increases proportionately to the amount of pumping done. Consequently, the pumps will operate entirely automatically without any special controlling apparatus, which is greatly in their favor, but the addition of the Cutler-Hammer controlling mechanism is considered a step toward greater economy.

The plant was designed and its construction superintended by W. K. Palmer, consulting engineer, Kansas City, to whom we are indebted for the foregoing information, and the pumps were supplied by the Worthington works of the International Steam Pump Co., New York.

* * *

The flexible staybolt problem is attracting much attention in railroad circles, and many meritorious devices have been brought before the public to reduce staybolt breakage. The boldest scheme we have yet seen is that of an inventor who proposed to replace the ordinary staybolt by a construction consisting of two eye-bolts connected by a link. One eye-bolt is screwed into the inner sheet and riveted over in the usual manner, but the other eye-bolt is made with a large collar and provided with a cap and nut. The cap is to cover a hole large enough for a man's hand, each staybolt hole being made thereby a handhole. While there can be but little doubt of the flexibility of the connection, it is open to the serious objection that in locomotive construction with staybolts placed as closely as is the usual practice, the handholes would cut out a large percentage of the outside plate, making it resemble the fluesheet rather than the ordinary boiler side sheet.

Copyright, 1905, by THE INDUSTRIAL PRESS.

Entered at the Post-Office in New York City as Second-Class Mail Matter.

MACHINERY

REGISTERED IN UNITED STATES PATENT OFFICE.

DESIGN—CONSTRUCTION—OPERATION.

PUBLISHED MONTHLY BY

THE INDUSTRIAL PRESS,

66-70 WEST BROADWAY, NEW YORK CITY.

LESTER G. FRENCH, Editor.
FRED E. ROGERS, Associate Editor.

The receipt of a subscription is acknowledged by sending the current issue. Remittances should be made to THE INDUSTRIAL PRESS, and not to the Editors. Money enclosed in letters is at the risk of the sender. Changes of address must reach us by the 15th to take effect on the following month; give old address as well as new. Single copies can be obtained through any newsdealer.

We solicit communications from practical men on subjects pertaining to machinery, for which the necessary illustrations will be made at our expense. All copy must reach us by the 5th of the month preceding publication.

JULY, 1905.

PAID CIRCULATION FOR JUNE, 1905,—22,563 COPIES
Including 600 advertisers' copies, as part of their contracts.

MACHINERY is published in four editions. The practical work of the shop is thoroughly covered in the Shop Edition—$1.00 a year, comprising more than 430 reading pages. The Engineering Edition—$2.00 a year—contains all the matter in the Shop Edition and about 250 pages a year of additional matter, which includes a complete review of mechanical literature, and forty-eight 6 x 9 data sheets filled with condensed data on machine design, engineering practice and shop work. The Foreign Edition, $3.00 a year, comprises the same matter as the Engineering. RAILWAY MACHINERY, $2.00 a year, is a special edition, including a variety of matter for railway shop work—same size as Engineering and same number of data sheets.

HOW MUCH DOES IT COST?

We publish this month a contribution upon estimating the cost to manufacture a small machine. While the amount of labor and material required to produce the parts of this particular machine is of no practical importance in itself, we consider the article of unusual value to the average machinist because it points out the method of procedure in making estimates upon machine work.

Ask a machinist at the lathe how long it will take to turn out a certain piece of work, or how much he would judge the piece to be worth, and in the majority of cases his reply will be wide of the mark. It is commonly said, and is probably true, that few machinists are able to make even approximately correct estimates of the cost of machine work. The reason for this is that the man who is regularly employed in performing routine duties seldom attempts to analyze his work or to observe how long a time the different operations require. He is content to perform his work in a mechanical way.

No written article can teach such a person the science of estimating. All that can be done is to show the method to be followed, and then the facts, data and conclusions must be derived through observation and experience by the person himself. In this article upon estimating, the author has pointed out the course of procedure and the reader must gather his own information according to the kind of work that he wishes to make estimates upon. We commend to all young mechanics who expect to rise above the position of an operative, the practice of observing and of accumulating data upon the cost of machine work. The time may come when the ability to form correct estimates of cost may mean promotion to a better position.

* * *

THE ETHICS OF CONTRIBUTIONS.

When a farmer "trades" a tub of butter at the local grocery, or a machine tool builder sells a lathe to the X Y Z Manufacturing Co., both part with their property and receive its supposed equivalent, in some form, but neither would sanely expect to sell the identical tub of butter, or the same lathe, to another customer. In the very nature of things, the carrying out of a multiple exchange of this order, or any attempt at it, would brand one as dishonest or crazy. Unfortunately, the same condition *per se* does not apply to selling the wares of a writer, but the ethics of contributing to publications *for pay* are the same. Why then, we wish to ask certain of our contributors, do you think that you can honestly send your contributions to MACHINERY, and to two or three other trade papers, and receive payment from each? You cannot, but nevertheless we have been put to considerable trouble of late by receiving contributions which were sent to us and in duplicate or triplicate to other papers, and the only excuse for the authors that we can imagine is that of ignorance. It is said that Myra Kelly sent her first contribution on East Side life in New York to both *McClure's* and the *Century* magazines, and when she found that the story had been accepted by both publications, she was, as you say, "in a pickle." On learning something of the rules governing contributions she also incidentally found out that there is a law designed to protect publishers from actions of this character, but of course no editor would prosecute an ignorant offender, although he might be severely tempted to make some unprintable remarks about the contributor's common sense. The sum and substance of the above scolding is, that when you offer your written wares to MACHINERY do not try to sell them to other customers until you know that we have rejected them.

* * *

VARIABLE SPEED MOTORS.

There is undoubtedly a general impression abroad that the usual speed range of an ordinary variable speed field-controlled electric motor is only about 2 to 1. In several articles published in recent issues of MACHINERY, this idea has been expressed, and correctly so, we believe, as regards the large majority of such motors. But, in justice to at least one concern, the Northern Electrical Manufacturing Company, Madison, Wis., it should be here stated that this limitation does not apply to the motors built by this company, inasmuch as while they are of the "ordinary" type they are made in variable speed ratios of 2 to 1, 3 to 1, 4 to 1, 5 to 1, and even 6 to 1. But, of course, the greater the speed variation the larger and heavier the motor for the same power output. For instance, a 10 H. P. Northern motor for continuous service, having a minimum and maximum speed of 650 and 1,300 R. P. M. weighs 1,235 pounds; when the speed variation is increased to 5 to 1, the same frame is required for a 4 H. P. motor. In other words, the greater the speed variation of any ordinary field-controlled motor for a certain power output, the greater must be the size of the motor. The same applies to a steam engine or any other mover in general; the engine which is required to deliver 100 H. P. to the line-shaft at 100 revolutions per minute must be practically twice as heavy as another which delivers 100 H. P. at 200 revolutions per minute. There is no doubt a great deal to be learned regarding electric motor construction, and no one in the present state of the art is prepared to predict just what changes will be brought about; but, in general, it may be said that this limitation must always apply—in a motor of the ordinary type—that the greater the speed variation the larger, heavier and more expensive must it be. It would, perhaps, be possible to build a variable speed motor which by field control alone would give a speed variation of 20 to 1; but its size and cost under the present conditions of building motors would be prohibitive.

* * *

The United States Consul at Prague, Austria, has had the enterprise to distribute a typewritten circular, and printed index cards for the purpose of securing data relative to manufacturers in this country and to their products, which he can have on file, at the consulate, for immediate reference, when inquiries are received by him. The index cards are called "Little Drummer" record cards, and to show their utility the circular gives the following rather concise and direct example:

"Mr. Buyer desires—well, say, automobiles. He addresses himself to this consulate; we have complete commercial data. He wants information immediately; we can furnish it immediately. We simply direct him to our "Merchandise File," which indicates to him the dealers, introduces the "Little Drummer" record cards, and drives in immediate sales. The information is most complete; he can cable an order immediately."

Mr. Ledoux, the consul, deserves much credit for his enterprise and should be backed up by those receiving the cards, to enable him to more completely carry out his mission as he sees it.

ENGINEERING REVIEW.

CURRENT EVENTS, TECHNICAL AND MECHANICAL—LEADING ARTICLES OF THE MONTH.

A recent report from Consul H. W. Harris, Mannheim, Germany, says that a newly-invented German typewriting machine is described in the German press. It is said to print syllables and short words instead of single letters, attains much greater speed than others, and, it is claimed, will revolutionize the art of typewriting. So far as can be ascertained the machine is not yet on the market. A leading newspaper comments that when it is remembered how many millions from Germany and all Europe have gone to America for typewriter supplies the new product of German invention will be greeted with satisfaction.

An extensive use of diamonds is made for dies for drawing wire of the smaller sizes, up to, say .025 inch in diameter. Diamonds are used for this work because the wear upon the hardest steel dies enlarges the hole in the die to such an extent that the diameter of the wire is not uniform from beginning to end of the drawing. With diamond dies it is practicable to draw platinum wire to a diameter of .0005 of an inch. The diamonds used for this work weigh from four to five carats each and cost from fifteen to twenty dollars per carat. It is said that several hundred thousand dollars' worth of diamonds are used as dies in the various wire factories in this country alone.

The U. S. Consul-General at Frankfort, Germany, refers in a recent consular report to an address delivered by Professor F. Linde before the Technical Society of Frankfort in which a process for making oxygen from liquid air was described. If liquid air is allowed to evaporate slowly, nitrogen evaporates first and then the oxygen; the liquid remaining after evaporation has proceeded for a certain length of time, therefore, contains a higher percentage of oxygen than the normal proportion. By repeating the process a liquid is obtained which contains about 71 per cent of the oxygen of the air, and rectification permits of a further increase of the percentage of oxygen. The cost of production of oxygen by the Linde process is said to be one cent for thirty-three cubic feet. A supplementary process makes it possible to obtain pure nitrogen from the atmosphere.

In an interesting paper on motor drive as applied to cotton mills, read by Mr. Meldon H. Merrill before the annual meeting of The New England Cotton Manufacturers' Association held at Boston, Mass., April 26-27, the author made out a very strong case as showing the relative economy of direct electric motor driving applied to textile machinery over the group driving system. As an example he quoted the spinning room in one of the large New England mills in which a 200- H. P. motor operates 52 ring spinning frames. The cost of the installation was $2,694.00, of which $2,060.00 is represented by the motor and $634.00 by the cost of belts and shafting. If these machines were direct connected to electric motors, 52 three-horse power motors would be required at a cost of $5,260, leaving a difference of $2,566 in favor of group drive. The interest on this amount at 5 per cent. would be $128.00 a year; but it was found from actual tests at the above mill that 37½ horse power was required to drive the shaft load, no frames being in operation. Taking into account the difference in efficiency of the large and small motors, this means that 18 horse power is saved by using the direct-connected motors, which figure at $25.00 per horse power per year amounts to $450.00 and which compared with $128.00, the interest on the increased investment, shows a saving of $322.00 a year as the result of direct-connected drive.

Consul-General W. R. Holloway at Halifax, Nova Scotia, says that the steel bridge now under construction over the St. Lawrence River at Quebec is a remarkable structure. The weight of this bridge will be about 35,000 tons. Its span of 1,800 feet crosses the entire St. Lawrence River at such a height as not to interfere with navigation, and will be the longest in the world, the Forth bridge in Scotland being 1,710 feet long, the Brooklyn bridge 1,680 feet, and the new East River bridge in New York 1,600 feet. There have been manufactured by the Phœnix Bridge Company, Phœnixville, Pa., to date, and partly shipped to the site of the bridge, about 10,000 tons of steel. It will take about two more years to complete the structure. The masonry piers are entirely finished, and the temporary false works, of steel, are now in place on the south shore, upon which erection will begin at the opening of spring this year. The 1,800 feet of steel-bridge work between the piers will be erected without any false work in the river. The bridge is to be 80 feet wide over all, carrying a double-track railroad, a double-track trolley and highway, and two sidewalks. Many novel features have been adopted in the design and manufacture of this bridge. The total length of the bridge will be 3,300 feet; length of channel span, 1,800 feet; ship clear headway, 150 feet above highest tide; height of cantilever towers, 360 feet above the river. The Phœnix Bridge Company are the contractors for the superstructure.

Electric welding by the well-known Thompson process is an American invention that is apparently attracting renewed attention in England, says the *Western Electrician*. A writer in the *Times* (London) notes that in many industries the system has caused a complete modification of existing methods of manufacture, and in some cases electric welding has created entirely new manufactures. The process is used for welding tramway rails, in the manufacture of steel chains, in the production of wire fencing and in the making of bolts, printers' chases, steel tubing, tires, hoops for casks, vehicle frames and many other articles. Electric welding has advantages which render practicable many operations which are impossible by the ordinary forge or gas blowpipe. Finished work may be welded and repaired without damage. The welding operation can be closely watched as it proceeds and faulty welds prevented. Welds are made with great rapidity, occupying only a few seconds, and in small work they are performed almost instantaneously. Impurities are excluded from the joint and a perfectly homogeneous weld is obtained. The results of tests of the comparative strength of electric and hand welds, carried out for the late Sir Frederick Bramwell, show that whereas in the case of hand-welded iron bars the ratio of weld to solid was 89.3 per cent., in the case of electrically welded bars the ratio was 91.9 per cent. The cost of fuel is, generally speaking, about the same for hand and electric welding, but with the electric process the labor is greatly reduced.

The tendency in the construction of large modern power plants is to concentrate everything under one roof, hence we have 50,000 and even 100,000 H. P. plants for the generation of electric power for railway operation. The supposed reason for concentration is economy, but there is undoubtedly a limit beyond which the mere aggregation of units ceases to show any additional economy. When a plant has reached such a size that the chief engineer needs two or three assistant head engineers to see that the machinery is being properly looked after and coal efficiently burned then that plant has exceeded the economical limit. In other words there is no gain in economy in building a power plant, all other things being equal, which exceeds in size that which one chief engineer can look after and keep personally in touch with all the employees therein. This point was brought out very clearly by Mr. George I. Rockwood in a paper on boiler plants read before the Boston meeting, April 26-27, of The New England Cotton Manufacturers' Association. Mr. Rockwood said that in order for one chief engineer to watch attentively the daily routine of the large power and boiler plant, it should be compact and of such shape and dimensions that the engineer can easily see

and speak to every fireman, coal passer and oiler without moving about much himself. The limit to a size of a boiler plant which one man can properly oversee in this way may be placed at something over 5,000 boiler H. P. Obviously no labor is saved by concentrating into one unit a larger plant than can be carefully attended to by a single engineer. This fact seems to have been overlooked completely by designers of the great metropolitan electric stations.

U. S. Consul Richard Guenther, at Frankfort, Germany, refers in a recent report to the German fear of publicity and particularly to a circular letter issued by the Chamber of Commerce of Sorau, Prussia, which was addressed to manufacturers and selling agents of that district. The letter is a remarkable plea for secrecy, advising manufacturers and agents to abstain from giving publicity to any details regarding their products except to those interested as customers. It urged that factories should be closed to foreigners or strangers and that all employes should be admonished to give nothing to newspapers or trade papers for publication. Mr. Guenther very properly closes with the following query:

"How do these German chambers of commerce, which represent the manufacturing and commercial elements of their country, reconcile this illiberal spirit with the fact that Germany sends individuals, official delegates, and even ministers of state to the United States to inspect our factories, trade schools, public institutions, etc., for the purpose of obtaining knowledge and benefitting German interests?"

PROTECTIVE DEVICES IN SHOPS.

Every year persons are killed or badly hurt in shops or factories by getting caught in moving machinery. It is, of course, practically impossible to prevent all accidents of this character, although a large proportion of them could be avoided by simple protective devices which cost very little, and give protection out of all measure to their cost. Our attention has been called to the danger that exists in many shops in the unprotected bars of stock running in turret lathes, by a distressing accident that occurred not long ago in a western machine shop. A long 2-inch bar with a rough crop end, was running in the machine, the outer part being held in the usual supports without a guard. A workman attempted to pass through the narrow space between the stock and a post with his back to the bar when the rough revolving end caught his jacket and threw him heavily to the floor, breaking his neck. While such accidents are very rare, there is a possibility of their occurrence wherever screw machine stock is running unprotected. It is a case where a simple pipe guard is not only a safety feature but an improvement in the support for long stock. A short opening between the end of the stock and the end of the pipe permits the stock to be manipulated, and so makes very little or no inconvenience. In the case of change gears for lathes, the danger of accident to life or limb is ever present if the gears are unprotected, and, from a merely selfish point of view on the part of owners, it is wise to cover such gears as they are liable to be broken by something else besides flesh and bone getting between them.

THE IMPORTANCE OF HAVING THE ENGINES OF TWIN-SCREW SHIPS WORK IN UNISON.

In the June, 1905, issue of MACHINERY an abstract was given of a paper read by Mr. A. Mallock before the Institution of Naval Architects in which the author outlined his method of reducing or entirely eliminating vibrations in twin-screw steamers. Briefly it consists of means by which the engines are made to run at the same speed but in opposite phase. In other words they are made to run so that when, say, the port low-pressure piston is moving through the down-stroke, the low-pressure piston of the starboard engine is traveling in the up-stroke. Strangely enough the author did not dwell upon another and equally important result obtained by his invention, and that is, making the engines run at the same rate. To the uninitiated it may seem of little moment if one engine of a steamship does run slightly faster or slower than its mate, but such is not the case. If one engine is running faster than the other it means that the rudder must be deflected to counteract the greater effort of that screw, and this means lost effort. The maximum efficiency on a straight course, other things being equal, can only be gotten when both engines are running at exactly the same rate. Hence the Mallock device, while no doubt highly beneficial so far as reducing vibration is concerned, should be of still greater moment as regards efficiency, in that keeping the engines in step the waste effort is reduced to a minimum.

MELTING STEEL WITH CAST IRON.

At the New York meeting of the American Foundrymen's Association, June 6-8, Mr. R. P. Cunningham read a paper on "Melting Steel with Cast Iron," in which he describes his method of securing strong homogeneous semi-steel castings. In his preliminary remarks he refers to the great increase of the strength of castings now required as compared with that of a few years ago; it was unusual, years ago, to receive an order for a pump to stand a pressure of more than 1,000 pounds per square inch, but to-day it is nothing uncommon to get an order for a pump to work under a pressure of 5,000 pounds and even higher. Stationary engine builders are called upon to build engines to work under 200 pounds steam pressure, whereas a few years ago 100 pounds pressure was considered the limit. The same condition exists in machine tool building, the speed of modern tools being nearly double that of a few years ago. Car wheels are required to carry loads 100 per cent greater than was required of them twenty-five years ago. Hence the problem of making strong castings is a very important one and is one that must be met by the successful foundrymen.

Mr. Cunningham's method of charging a cupola is as follows: If a casting is wanted that requires 4,000 pounds of metal with 25 per cent of steel, a 48-inch cupola, say, is charged on the bed with 1,200 pounds of coke; on top of this coke is put 1,000 pounds of iron, then 500 pounds of soft steel scrap, then 500 pounds of iron, then 150 pounds of coke, 500 pounds of steel, 1,500 pounds of iron. With the last amount of steel 1¼ pounds of ferro-manganese should be added for every 100 pounds of steel used, and the same amount of ferro-silicon is put into the ladle. The results of 18 casts with different percentages of steel have apparently demonstrated that the highest amount of steel that can be used to advantage is 33 per cent. Higher percentages of steel than 33 per cent cause excessive shrinkage and only slight gain in strength. The highest tensile value reached with the test bars was 33,205 pounds and the lowest was 31,890 pounds for the perfect bars. The highest transverse strength was 3,335 pounds; the lowest for a perfect bar, 3,180 pounds. For ordinary work 25 per cent steel will give sufficient strength for all practical purposes and will make castings that machine easily and are close-grained.

BALANCING CENTRIFUGAL PUMPS.

A paper, "Some Types of Centrifugal Pumps," by Mr. Wm. O. Webber, read at the Scranton meeting of the A. S. M. E., gives something of the history of centrifugal pumps and reviews some of the improvements that have been made up to the present time.

The early centrifugal pumps had a single impeller, or runner; some of them had the suction on one side only and others had double suctions, or openings, on both sides. These early pumps had low efficiency. From 46 to 60 per cent seems to have been the best results obtainable under heads varying from 4½ to 15 feet; 40 feet was formerly considered the maximum height at which centrifugal pumps could operate efficiently. Sulzer, of Winterthur, is credited with having been the first engineer to discover the value of compounding centrifugal pumps. Following him pretty closely are, A. C. E. Rateau, of Paris, France, and John Richards and Byron Jackson, of San Francisco, California. All the pumps designed by these men were constructed originally with the impellers facing in one direction; the consequent thrust on the impellers was partially taken up by balancing chambers upon the rear of each impeller. The residual thrust was further taken care of in Rateau's pump by a balancing cylinder, shown at the left of Fig. 1, into which water was admitted from the dis-

charge end of the pump. A similar arrangement was used in Richards' pump, shown at the right at *C* in Fig. 2.

Rateau further tried to improve the balancing of his pump by constructing enclosed impellers having one of the side plates of the impeller of smaller diameter than the other, so that the areas of the two impellers' side plates, subjected to the discharge pressure from the impeller, were equal. Sulzer then made a greater improvement in balancing by placing the impellers back to back in pairs, so that the thrust of one impeller was counterbalanced by that of the other. This method, however, required short tortuous distances between the pump chambers. Richards followed the same plan, but utilized it in a two-stage pump only.

Mr. Webber has designed a balanced compound centrifugal pump, in which the impellers are placed back to back, but not in pairs serially disposed, as will be noted in Fig. 3. In this design the sum and difference of the suctions of the discharge pressure of all the impellers balance each other. The Webber compound centrifugal pump has been built for heads of 1,200 feet, in which case two 4-stage pumps coupled in series are employed.

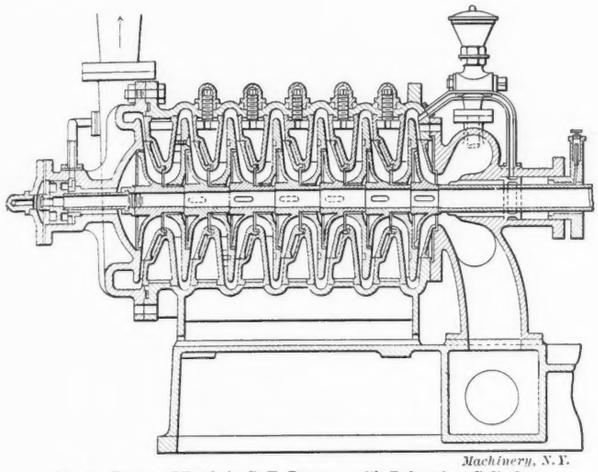


Fig. 1. Pump of Prof. A. C. E. Rateau, with Balancing Cylinder.

In connection with the foregoing mention may be made of a recent improvement in balancing centrifugal pumps patented by E. S. Lea and J. Degen, of Trenton, N. J., to which our attention has been called by Mr. Fred W. Barnaclo, of Francis H. Richards' office, this city. This invention is shown in sectional view Fig. 4. The impellers in this pump are not placed back to back but follow the earlier forms of construction. The balancing feature is said to work automatically and to require no equalizing piston whatever. The action of the fluid flowing from the mouths of the impellers creates a partial vacuum

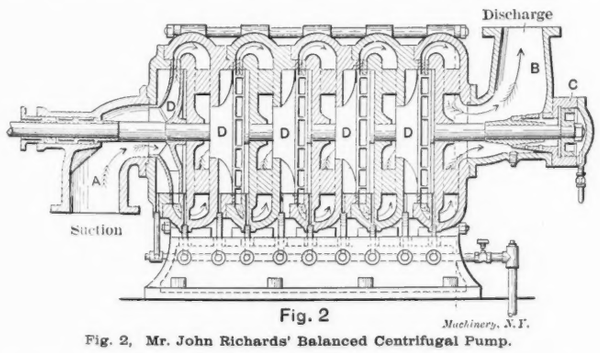


Fig. 2, Mr. John Richards' Balanced Centrifugal Pump.

in the chambers *A* and *B* which, of course, reduces the pressure in these chambers. In order for the partial vacuum to be equal on both sides of the impellers, it is necessary for the distances between the rim of the impeller *D*, and the walls, *C* and *E*, to be equal. Should these distances on each side of the impellers be made unequal by shifting them axially, it will follow that the suction will also be unequal on each side of the impellers, and consequently the reduction of pressure would also be unequal. Hence the normal action of the impellers causes them to assume a central position.

THE REAL EFFICIENCY OF STEAM BOILERS.

W. H. Booth, in Electrical Review (London), May 26, 1905.

In calculating the efficiency of a steam boiler the customary method is to find the ratio between the number of thermal units contained in the steam produced by the boiler and the calorific capacity of the fuel burned to produce that result.

The calorific capacity of a fuel is usually stated as the number of thermal units that it will produce when burned, and it is assumed that if there is hydrogen in the fuel the water

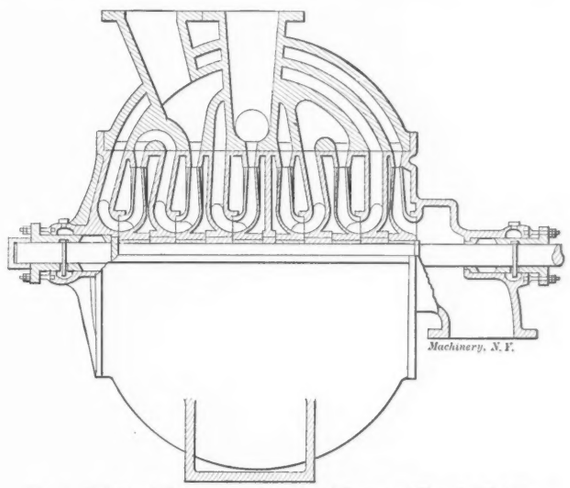


Fig. 3. Balanced Compound Centrifugal Pump of Wm. O. Webber.

produced by this hydrogen shall be assumed to be finally measured as water, and not as steam, at the initial temperature of the atmosphere to which the products of combustion are supposedly returned. Theoretically this is correct enough indeed, considering that the hydrogen in the coal is in solid form, whatever the chemical composition of such solid may be, it would appear to be fully as correct to base the calculation on the supposed return of the products of combustion to the

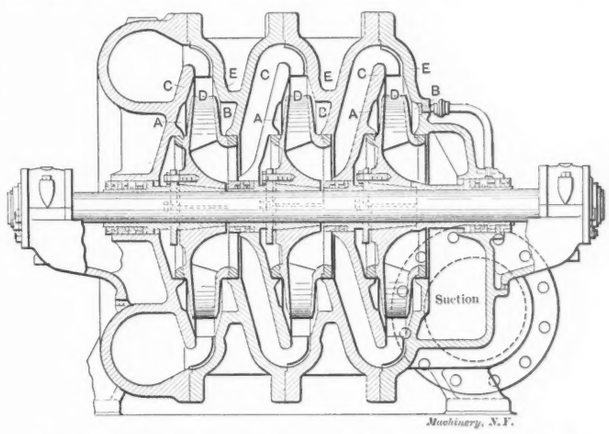


Fig. 4. Improvement in Balancing Centrifugal Pumps of E. S. Lea and J. Degan.

state of a solid. However, assuming the ordinary practice, it seems scarcely correct that a boiler should be debited with all the thermal units produced by the hydrogen when it is impossible that the water produced shall be reduced to liquid form. Occasionally in flue feed heaters or economizers, the water, or some of it, is condensed on the colder lower ends of the water pipes, and combining with the products of the combustion of the sulphurous parts of the fuel, produces very rapid and serious destructive corrosion of the external parts of these pipes, so that in practice good maintenance involves as a *sine quâ non* that the water shall not be asked to give up its latent heat, and it is found not to do so unless the surfaces with which it comes in contact, fall below about 100 degrees F. The true calorific capacity of a bituminous or hydrogenous fuel ought therefore to be calculated on the basis of 52,290 B. T. U. per pound of hydrogen, in place of the customary 62,100 B. T. U.

Since no boiler works below 100 degrees F., this correction

from customary methods ought properly to be made in the calorific value of a fuel, but it should be stated, of course, in any test figures, since the use of the incorrect figure is so firmly established, and the newer figure is just the sort of thing that the company promoter's expert would be likely to fasten upon to embellish his results, much as he now will sometimes innocently state the calorific capacity of fuel in B. T. U. without troubling to mention that he means per kilogram, or *vice versa*, if the other way around better suits the desired end.

But even the above method of stating fuel capacity is not right, and it tells against the high-pressure as compared with the low-pressure boiler, which is so much cooler, and is therefore better able to absorb lower temperature heat.

It is an obvious impossibility for a steam boiler producing steam at a temperature of 290 degrees F. to reduce the temperature of the waste gases below 290 degrees F. It would require an infinite area of heat-absorbing surface to effect even this result, but it can at least be stated that the temperature of the boiler itself is the minimum to which furnace gases can be reduced by the boiler unaided by a lower stage of feed heating in a separate vessel. Seeing, then, that a boiler cannot absorb the heat below 290 degrees F. in the suppositious case, is it reasonable to base the efficiency coefficient of a boiler upon the amount of heat supplied to it at any temperature? It would appear to be more correct in every way to debit the boiler with all the heat supplied to it above its own temperature, which is, of course, that of the saturated steam within it.

Thus, let it be assumed that a fuel consists of pure carbon giving 14,647 B. T. U. per pound, what is the efficiency of a steam boiler which produces per pound of fuel, only 10 pounds of steam at 174 pounds absolute pressure from feed water fully heated to 370 degrees F., which is the boiler temperature? The steam contains 8,521 B. T. U. of latent heat, and the customary efficiency of the boiler would be $\frac{8521}{14647} = 58.1$ per cent.

The furnace products weigh, we will say, 22 pounds per pound of fuel, and the boiler temperature is 370 degrees, while the external atmospheric temperature is 40 degrees F. Then the furnace gas is carrying away, or, rather, must carry away not less than 330 degrees of temperature, and its specific heat is 0.24. Then $22 \times 0.24 = 330 = 1{,}722$ B. T. U. The maximum available calorific capacity of the fuel is therefore only $14{,}647 - 1{,}722 = 12{,}925$ units, and the actual efficiency of the boiler is $\frac{8521}{12925} = 66$ per cent nearly, or 8 per cent more than shown by ordinary methods. However large the boiler, it could not possibly have absorbed more than 12,925 units. The possible performance is, in fact, only a full 88 per cent of the fuel capacity, and as no boiler is likely to reject gases at less than 100 degrees above its own temperature, the practical efficiencies are much less than present crude methods would indicate.

As an example, let the above number of units, 1,722, be subtracted from Shireoaks coal which has recently been stated to possess 11,886.7 B. T. U., and to be associated with an efficiency of over 84 per cent. We have $11{,}886.7 - 1{,}722 = 10{,}164$ as the maximum heat available. Now if 11,886.7 units show 84 per cent efficiency, the boiler must have absorbed 9,985 B. T. U. But $9{,}985 \div 10{,}164 =$ true efficiency of boiler $= 98.2$ per cent. This figure shows how very easy it would be to obtain a real efficiency of heat absorption of over 100 per cent by making an error in the value of the fuel. It also shows how very serious may be the effect of excess of air in high-pressure boilers, for an excess of air raises the number of heat units which lie below the datum line of possible heat absorption, not to name that of practicable heat absorption. It is much to be desired that boiler efficiencies should be based on the correct available heat value of the fuel, that is, on the heat supplied above boiler temperature. It may be argued that this brings the furnace into the account. So it does, but it also serves to draw attention to the harm done by inefficient combustion and excess of air, and if furnaces and boilers are to be rated as one, it would tend to better furnace practice. Boiler efficiencies would all appear higher than they do now, but some of the present high efficiencies claimed would not bear investigation.

REINFORCED CONCRETE.

Engineering, London, April 28, 1905.

Up to the last few years the use of concrete as a building material was chiefly confined to the construction of foundations, piers, reservoir dams, and similar purposes, in which the stresses to be met were almost entirely simple pressures. Indeed, even fifteen years ago many engineers looked askance on the use of concrete for arches, considering it for this purpose much inferior to brick. Much of the caution shown in extending the uses of this valuable material doubtless arose from the frequency with which concrete masonry exhibited unsightly cracks, due largely to the material being allowed to get dry while hardening. At the same time, careful examination has shown that cracks of the same character are common in masonry of all kinds, but are unnoticed, because they follow the regular joints of the structure; whereas on the smooth uniform surface of concrete, cracks of much less significance are immediately visible.

The plan of reinforcing the material with metal, of which several systems have been introduced during the last four years, has greatly extended the possible use of concrete; and it appears that in many cases a reinforced concrete bridge may compete, even in first cost, with a steel girder; while as regards upkeep, it has, of course, many advantages. Small bridge culverts of this material were extensively used by Russian engineers in building the Manchurian Railway. For openings of some 7 feet span, flat slabs of concrete reinforced with rails were used, the thickness being 8¼ inches. A similar system was used for spans up to 21 feet, the concrete, however, being thickened at the center as the span increased, the depth at this point being 2 feet 6¼ inches for the 21-foot span, and proportionately less for smaller openings. The thickness at the bearings was, however, the same in all cases, viz., 8¼ inches. The line was thrown over the spans as little as seven days after completion. The concrete consisted of one part cement, two sand, and five broken stone. The system in this case had great advantages, as stone for masonry was unobtainable, and could, moreover, only be used for arches, which would have necessitated the use of higher embankments than were required with the ferro-concrete used as described. Much larger spans have, of course, been built than those mentioned. One of 153 feet span, carrying four main-line tracks, has recently been built for the Lake Shore and Michigan Southern Railroad, while Mr. Edwin Thacker, M. Am. Soc. C. E., states he considers the system feasible for spans up to 500 feet, and has actually got out designs for a span of 300 feet, the cost comprising favorably with that of a steel bridge.

One great drawback to the extension of the system lies in the difficulty of proportioning structures thus built in a thoroughly rational manner. In the case of steel bridges, certain simple assumptions as to the elasticity and strength of the material suffice. These assumptions are doubtless not absolutely exact, but are sufficiently near the truth for practical purposes. The elastic properties of concrete are, however, very different from those of steel; Hooke's law is not even approximately correct, and, moreover, the material always takes a permanent set when first loaded. The true distribution of the stress and strain on a concrete beam is thus a much more complicated matter than it is in the case of a steel joist, in which it is permissible, within working limits of stress, to assume the accuracy of Hooke's law. The assumption generally made in the case of ferro-concrete is that plane sections of a concrete beam remain plane after bending. This postulate is, of course, that commonly made in proportioning steel work; and in the latter case, stress being proportional to strain, the usual formula for the working strength of beams is readily deduced. In the case of concrete, however, the stress-strain curve is much more complex.

In view of the uncertainties introduced by the different factors above mentioned, it is really questionable whether, after all, the theoretically objectionable formula of M. Henne-

bique is not as good as any other. The latter all involve a preliminary calculation of the position of the neutral axis, which varies with the percentage of metal used, and with the type of stress-strain curve assumed for the concrete; and also with the maximum stress at any particular section. Thus, in a centrally loaded beam its position at the ends is entirely different from what it is at the center. M. Hennebique, on the other hand, makes no attempt to locate this neutral axis, and simply assumes that one-half of his beam resists compression, and that the stress is uniformly distributed over this half. The moment of this compression about the center of the section he equates to half the moment due to the load, and the other half of the moment due to the load he equates to the moment about the center of the section of the tensile stress on the metal reinforcement. The working strength of concrete in compression he takes as 350 pounds per square inch, and neglects entirely its strength in tension. The working tensile stress on the steel reinforcement he takes as 14,000 pounds per square inch. The method is, of course, totally illogical, yet many thousand cubic yards of ferro-concrete have been successfully designed on these lines; and a comparison of the strength of the ferro-concrete beams as calculated by this formula, and by those of a more rational type, shows very little difference between the two for a considerable range in the ratio of metal to concrete. On the other hand, it must not be forgotten that formulæ, which are non-rational in form, are always risky when applied to extreme conditions.

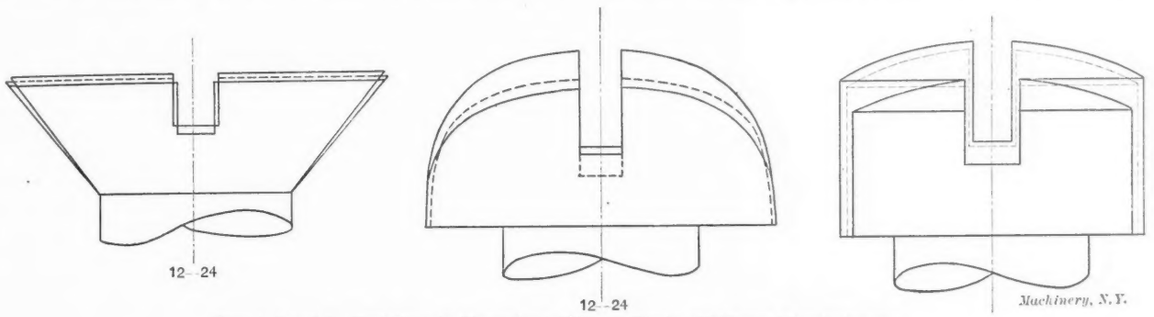
Concrete being as weak in shear as in tension, provision is also required to take the shearing stresses. Some American designers have to this end patented special forms of reinforcement bar, in which each main tension bar has projecting upward from it ties inclined at an angle of 45 degrees. These extend to the top of the bar and take the tensile stresses arising from the shear. The corresponding compressive stress at right angles to this is carried by the concrete. The system is doubtless efficient, and on large spans, where weight must be reduced to a minimum, it may have some advantage; but in work of ordinary proportions it seems to be little superior to the Hennebique system, in which the necessary strengthening is provided by stirrups of flat iron bent into a U shape. The main reinforcing bars rest in these stirrups at the lower ends. The spacing of the stirrups depends upon the "web stresses" to be taken, which can easily be calculated by assuming the reinforced beam to be a latticed girder, the lower chord of which is represented by the metal reinforcement, the upper one by the center of the compression half of the beam, while the stirrups represent vertical ties, which may be taken as connected together at top and bottom by inclined imaginary struts. The advantage of this simple method of reinforcing for shear lies in the possibility of using common rolled sections for the whole of the reinforcement.

Occasionally doubts have been expressed as to whether the metallic reinforcement may not suffer from corrosion as time goes on. This would be extremely dangerous if it occurred, since the metal being out of sight, its loss of strength might remain undetected until, some day, the structure might fail under its ordinary working load. Fortunately, much evidence is available to the effect that steel or iron thoroughly embedded in concrete are permanntly protected from rust. Our American friends, indeed, are so positive on this point that they have recently constructed a number of reservoir dams in ferro-concrete. In some cases these have been arched, but in others they have been straight. The cross-section in the latter case is generally a hollow triangle, the sides of which are connected together by diaphragm walls from point to point. The dam is also anchored to its site, though generally the weight provided is sufficient to make the structure safe against over-turning, quite apart from the help received from the anchor-bars.

NOTE ON HEADS OF MACHINE SCREWS.

Paper by H. G. Reist, read before the Scranton Meeting of the A. S. M. E.

The size and shape of machine screw heads as made by different manufacturers vary greatly. In the case of screws made by some manufacturers the heads used on different sizes have different proportions, thus presenting a different appearance. The company with which I am connected is a large consumer of machine screws, and this variation has from time to time caused great inconvenience. Frequently designs are made to suit the screw heads of one manufacturer, and if afterward it becomes desirable to purchase screws manufactured elsewhere, there may be an interference with other parts of the machine, due to the variation in size of heads.



Figs. 1, 2 and 3. Variations in Dimensions of Screw Heads of Different Manufacturers.

When fillister head screws were used with the space for the heads counterbored, it was necessary to make the counterbore large enough to take the maximum size of head, thus not making a good appearance when smaller heads were used. The same was true in the case of flat-headed machine screws. The subject was taken up with several manufacturers of machine screws and it was found that they had no defined standard, but were open to suggestions, and were willing to supply screw heads to any ordinary dimensions without much if any extra cost. The extreme variation in dimensions found is shown graphically in the accompanying drawings by full lines, from which it will be seen that there is great irregularity, and, as stated above, the same manufacturers will in some cases supply larger heads proportionally than others. (See Figs. 1, 2 and 3.)

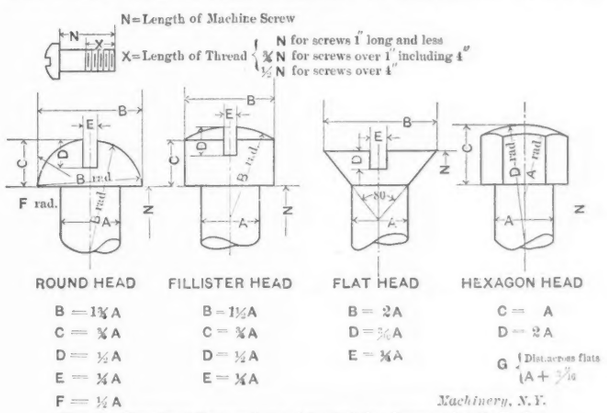

N=Length of Machine Screw
X=Length of Thread { N for screws 1" long and less; ½ N for screws over 1" including 4"; ½ N for screws over 4"
ROUND HEAD
B = 1⅝ A
C = ¾ A
D = ½ A
E = ¼ A
F = ¼ A
FILLISTER HEAD
B = 1½ A
C = ¾ A
D = ½ A
E = ¼ A
FLAT HEAD
B = 2A
D = ⅝ A
E = ¼ A
HEXAGON HEAD
C = A
D = 2A
G { Distance across flats (A + 3/16
Machinery, N.Y.

Fig. 4. Standard Proportions Adopted for Screws by G. E. Co.

In order to establish uniformity in our own works, we prepared the accompanying formula and table giving the shapes and dimensions of the heads of the size of screws ordinarily used. The shape and proportional dimensions of these heads are shown by the broken lines in the drawings of the several forms of screw heads. It will be seen that these heads pre-

sent a uniform appearance, and by submitting the table of dimensions we have had no difficulty in obtaining screws to the dimensions shown—in fact, the manufacturers have expressed themselves as pleased to have a standard to work to.

The author understands that there is a committee (appointed by the society) at work on the standardization of machine screw threads, and it would seem to him that a report from them along these lines might make a valuable appendix to their work.

SMOKE AND ITS ABATEMENT.

Abstract of Paper by Prof. Chas. H. Benjamin, read before the Scranton Meeting of the A. S. M. E.

Objectionable black smoke is due to the presence of hydrocarbons in the fuel and is produced as follows: The hydrogen and carbon compounds in the coal are driven off as gas by the heat at a comparatively low temperature, and may escape unburned. In this condition they would not constitute smoke in the common sense of the term. If heated to a sufficiently high temperature in the presence of air they burn with a yellow flame. If the air supply is insufficient, poorly mixed with gas, or if the temperature is lowered in any way, combustion is checked and carbon is deposited in the form of soot, or carried off with the gas as smoke.

Only three conditions are necessary for complete combustion, the proper temperature, the proper air supply, a thorough mixing of the air and the hydrocarbons.

The last condition is as important as any and is one too often neglected.

It is this condition which gives the gas or liquid hydrocarbon an advantage over the solid, since the atomizing of the former by the steam or air jet insures the most intimate contact between the air and the fuel.

The use of pulverized coal in combination with air or steam is a close approximation to the above, and, when properly managed, gives good combustion, no smoke and a high efficiency. The cost of pulverizing and the impracticability of storing pulverized fuel have, so far, hindered the more general adoption of this process, except for metallurgical work. When coal in the ordinary form is used as a fuel, smoke abatement involves some means of varying the coal supply and the air supply according to the demands made upon the boiler. When ordinary hand firing is resorted to, the great irregularity of the coal supply will cause poor combustion and smoke unless the air supply is varied to correspond. Steam jets are frequently employed under these circumstances, and, if properly put in, will improve the combustion by drawing in additional air over the grate and mixing it with the products of combustion in front of the bridge wall. The steam jet should be semi-automatic, the steam and air being turned on by the opening of the fire door and gradually closed off by a dash-pot attachment.

The best solution of the smoke problem, so far, has come from the introduction of mechanical means of handling the coal, which give a uniform feed to the fuel and a corresponding delivery of air for combustion.

The principle of the inclined grate stoker, as exemplified in the Wilkinson, Brightman and Roney stokers, involves the slow coking of the coal on a dead-plate, the pushing forward onto the top of the incline and the gradual descent, impelled by oscillation of the grate bars, until the combustion has left nothing but ash and clinker at the bottom.

When used with a fuel which does not cake or clinker too much and when not crowded too hard these stokers are economical and reduce the smoke considerably. If, however, it becomes necessary to slice and poke the fire on account of caking coal or overcrowded boilers, unburned masses of coal are rolled to the bottom and holes are made in the fire through which cold air rushes. Both of these circumstances make for poor combustion and a smoky fire. As a rule firemen poke the fire on stokers too much, doing more harm than good.

From observations covering a period of several years I have come to the conclusion that the traveling or chain grate is the best one yet devised for abating smoke. The grate used in the Babcock & Wilcox and Green stokers is of this type. To prevent waste through the grate the latter is usually quite close and it is necessary to use more draft than with an ordinary grate. A damper is used under the grate to prevent an excess of air passing up behind the grate or through the comparatively bare grate at the rear end.

Underfeed stokers operate on an entirely different principle, the coal being fed in underneath the grate and forced up through a rectangular opening in the center. A forced blast is used and the air for combustion is blown up through the coal, the tuyeres being on either side of the rectangular opening just mentioned. By this arrangement the fresh coal is always underneath and the distilled gases are obliged to pass through an incandescent mass of fuel in company with the air. With a proper pressure of blast perfect combustion is thus almost unavoidable. The ash and clinker are now at the top of the fuel, which forms a gradually rising mound in the center and pushes the clinker over to either side, whence it is removed by hooks through doors at the front. The heat generated is such that the ash generally melts and forms a sheet of clinker which can be easily removed without disturbing the fire.

I believe the automatic stoker is the most economical solution of the smoke problem. In forming this opinion I do not rely upon expert tests for efficiency; there are so many variables entering into the question that it is difficult to make accurate comparisons in this way. As it is entirely possible to improve the efficiency of a furnace ten or fifteen per cent by intelligent hand firing, there is much difficulty in determining the actual saving effected by a stoker. During a competitive test between the hand-fired furnace and the stoker the conditions are often entirely different from those obtaining in every-day use, and these changed conditions are usually more to the advantage of the hand firing.

By this I mean that good hand firing is the exception rather than the rule, on account of the dirty, disagreeable nature of the work and the low grade of help employed. With the introduction of mechanical handling there is a reduction in the quantity of manual labor and there should be an improvement in its quality. Manufacturers should understand that if they expect to get the benefit of improved machinery they must have men competent to run it to the best advantage.

The surest method of determining the relative economy of the two methods is by comparing the coal bills before and after the change, making due allowance for any variation in the work done. This has been done in a number of instances to my knowledge and the result has always been favorable to the stoker.

RECORDING ANGULAR VELOCITIES OF MARINE SHAFTING.

John H. Heck, Institution of Naval Architects, April, 1905.

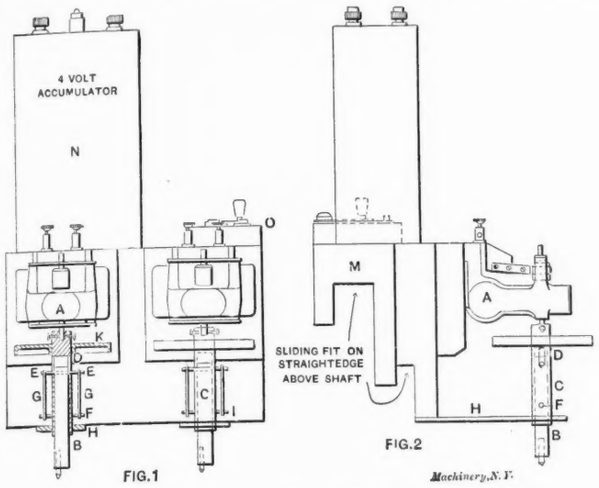
It has often been assumed in investigations on the balancing of engines which have appeared in papers read before technical societies, that the angular velocity of the shafting was uniform or nearly constant. Some tests carried out by the author of the paper here abstracted, show that such is often not the case.

The method successfully employed for the determination and recording of the variations of shaft velocity consisted of a small vertical electrical motor, which was kept constantly revolving at a high speed by means of a storage battery. At the lower end of the spindle of the motor, a brass sleeve containing a metal pencil-holder was fixed; this contained a lead pencil which was fitted eccentrically so that a circle about 5/16 inch in diameter was traced when the spindle revolved. The arrangement of this apparatus is shown in Figs. 1 and 2, in which two motors are shown to be used for checking purposes, although in a large part of the trials only one motor was used.

The motor was fixed to a small teak frame, which was so made that it could slide along a teak batten about six or seven feet long; this batten was fixed in the shafting tunnel in a fore and aft direction, by means of two special screw clamps clipping the angle-iron stiffeners on the tunnel sides in such a manner that the center of the spindle of the motor was directly over the longitudinal center line of the shafting, and the pencil point forced slightly upward, thereby extending a small spring by the tension of which the pencil-point was kept in contact with the paper on which the curves were

traced. This arrangement of the instrument in the shaft tunnel is shown in Fig. 3. For taking diagrams of velocity, a number of sheets of drawing paper were wrapped around the shaft and kept in position by elastic string.

The low-pressure crankpin was generally placed on the top center, the instrument started by switching on the current, and the motor frame pushed by hand slowly along the teak batten or straight edge, when the pencil traced a looped curve in the fore and aft direction; a line drawn through the center of the curve was taken as a datum line, and marked "L. P. crank on top"; the forward and after edges of the sheets



Figs. 1 and 2. Apparatus for Recording Velocities of Shafting.

of paper were also respectively marked "forward" and "aft." The instrument was then either tilted or taken off, so that the lead pencil was clear of the paper.

When the engines were running at various speeds, the curves were obtained by bringing the pencil in contact with the paper, and starting the motor, which, when fully in motion, was pushed slowly along the batten by hand, the pencil thus describing a continuous screw thread of loop curves on the paper which enveloped the shaft. Fig. 4 shows some of the actual curves as they appeared when the paper bands were taken off the shafts. The distance between each small curve from crest to crest, represents the distance passed over by a point in the circumference of the shaft during practically a very small uniform interval of a second. By counting the number of loops in one series all the way around the shaft,

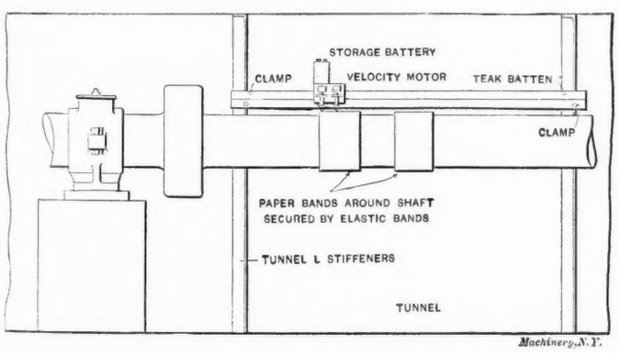


Fig. 3. Arrangement of Instrument in Shaft Tunnel.

the number of revolutions of the motor for one revolution of the engine was determined, and by measuring the distance between the crests of the small curves and dividing this distance by the radius of the shaft, the angle in radians passed through by the crank could be determined.

Curves were taken in a number of vessels when the machinery was tested at sea, and also at the mooring trials. In a few cases where it was possible to test the engines under somewhat similar conditions, it was found that at certain revolutions the variation of angular velocity per revolution was less when the vessel was moored than when she was under way at sea.

Fig. 5 gives the recorded variation of angular velocity in one case, the points of the diagram being calculated from the curves obtained as described, and connected by straight lines. The steamship on which this particular record was taken was a moderate size vessel fitted with two-crank compound engines working at eighty pounds pressure; weather fine; propeller well immersed. Comparing one ⅛ of a revolution with another ⅛ of the same revolution, the variation in the angular velocity of the shafting at 58 revolutions was 12 per cent, while making the same comparison with ¼ of a revolution the variation was 8.6 per cent. In this connection it may be said that the tests seemed to show that two-crank engines, even though well arranged and designed, are not so efficient for marine service as engines with a greater number of cranks. A large steamer fitted with three-crank triple expansion engines, working at 180 pounds pressure, for example, varied in the angular velocity of the shafting at 66 revolutions only 4.6 per cent, comparing one ⅛ of a revolution with another

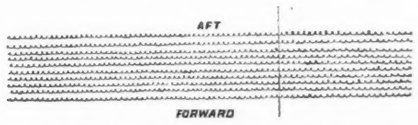


Fig. 4. Set of Curves Taken. Vertical Line is the Datum Line or L. P. Crank on Top.

⅛ of the same revolution; and at 54 revolutions 5.5 per cent. Making the same comparison with ¼ of a revolution, the variation when running a mile at 66 revolutions, was 2.1 per cent.

As an illustration of the effect of slight racing on the variation of angular velocity of shafting, in the case of a large three-crank triple expansion engine steamship (table of tests not reproduced); when no racing was apparent, comparing one ⅛ of a revolution with another ⅛ of the same revolution, the average variation of the velocity in the shafting at 68 revolutions was 7 to 9 per cent, while, when racing was evident, the same comparison showed a variation in velocity of 21 per cent. In other words, while the average number of revolutions of the engine was 68 per minute, the engines actually, during a part of some of the revolutions, were going at the rate of 82 revolutions, while, during a part of other revolutions, they were going at the rate of 58 revolutions per minute. It therefore may be said that if similar tests were made at sea while the engines were racing heavily, the acceleration at intervals in the rate of revolutions per minute

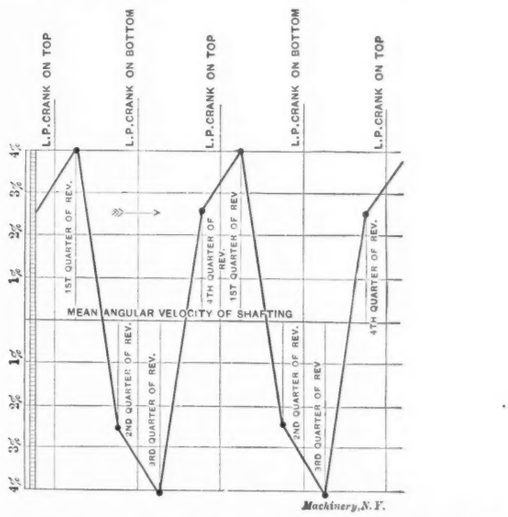


Fig. 5. Variation of Angular Velocity on Steamship having Two-crank Compound Engines.

would be found to be quite sufficient to account for the signs of strain which have appeared in the machinery of vessels after encountering very bad weather.

The variation in the angular velocity per revolution was nearly always greatest at low speeds of revolution, and as the revolutions became greater the variation became less. When an exception to this rule took place vibration became apparent. The variation in angular velocity increased with the vibration increase. The ship when vibrating appeared to

act as a powerful brake on the engines, which was applied hard at some intervals of the revolutions and released at others. Up to a certain number of revolutions of a ship having triple expansion engines, tested, the variation of the velocity in the shafting was normal. At 82 revolutions, however, the vibration was considerable, and on comparing one ⅛ of a revolution with another ⅛ of the same revolution, the velocity was found to vary from about 12.5 per cent above to 12.5 per cent below the mean angular velocity of the shafting.

CAN A STEAM TURBINE BE STARTED IN AN EMERGENCY QUICKER THAN A RECIPROCATING ENGINE OF THE SAME POWER.

Paper by A. S. Mann, Read before the Scranton Meeting of the A. S. M. E.

If a large steam turbine is cold and at rest how quickly can it be started? Can it be brought up to speed as readily as can a good cross compound engine that is cold all over?

Most station men would have doubts as to the adaptability of the large turbine, say 1,500 kilowatts or 2,000 horse power, for emergency work. So much has been written about the sensitiveness of a rotating disk to the changes of temperature and the effects of unequal expansion that it is easy to imagine difficulties in the rapid start. The possibilities of an engine with a 62-inch low-pressure cylinder in starting practically cold and coming up to synchronous speed are well understood. A station manager would criticise an engineer who would open his throttle as fast as he dared without wrecking his piping system and let his machine jump into her work. One turn at a time on the throttle is about all that is considered safe, and even then a close watch is kept for groaning valves and cold back bonnets.

Every time the starting valve is moved to increase the steam flow the engine is allowed to take its full increment of speed, due to that particular throttle position before the supply valve is moved a second time. There are ten large oil cups, and frequently more, that must be opened and adjusted before the machine moves at all, beside whatever oiling is to be done about the air pumps and other auxiliary apparatus.

Most engineers would consider ten minutes as rather a fast start and fifteen minutes as a more usual starting period, including time taken for warming up; in fact, it may not be overstating the case to say that if it were known that an engine-driven plant were to be called upon in emergency for power and it were essential that the briefest possible time were to elapse between the call and the taking of the load, one or more engines would be kept in motion all the time, turning slowly and hot all over.

This question makes itself very prominent when the steam station is operated as an auxiliary to a large source of high tension power, which is itself in the construction stage and has a large overload capacity of its own to carry, supplying all sorts of apparatus that use electric power, railway, lighting and power circuits simultaneously. At such a time all sorts of accidents will happen to the high-tension water-driven plant, most of them due to the necessarily temporary character of many of the electrical connections. It takes months before an intricate system of wiring can be thoroughly relied upon, for it takes months before the temporary work of construction can be replaced.

The station at present under consideration is equipped with three Curtis turbine-driven alternators, 40 cycle, 10,000 volts, each of 1,500 kilowatts normal capacity. During the summer months the station is operated as an auxiliary to a water-power plant, taking all sudden overloads. A signal has been arranged, a ¾-inch whistle, so that it can be blown instantly should the power fail. A blast of that whistle means—cut in two turbines and bring the third up to speed. The load will be heavy, and all auxiliary apparatus must be in regular operation. Each turbine has a surface condenser and there are three or four pumps to be started for each pair of turbines; one circulating pump, one combined hot-well and feed pump, one pressure pump for the step bearings and one dry air pump, all of which are motor driven. The exciter is driven by a steam engine and must be started also, for it supplies current to a portion of the auxiliary apparatus.

The boiler room has steam up at all times, supplying a system for manufacturing purposes other than power, and slow fires are kept in enough boilers to make steam needed for the normal load. Forced load means forced fires. The boilers have underfeed stokers, equipped with pressure blast, and will respond quickly to a 50 per cent excess call for steam. The operating force for this is about equivalent to a force for an engine-driven plant. Engineers and oilers, however, are busy about the building on construction work, installing new apparatus and taking such work as their regular occupation when the turbines are not running.

At the sound of the whistle the water-tender starts a blower on the extra row of boilers; all blast dampers are opened up and all stokers are allowed to feed at the maximum rate. Each fireman dumps his free ash and bars over his red fire. The man in charge of the coal and ash conveyor starts the pressure pump for step bearings. One of the turbine men starts the exciter which supplies current to the auxiliaries beside its field current; a second turbine man starts the circulating pump and then his turbine. The hot-well pump and the air pump are started by the oiler. These movements take place simultaneously. The force is organized upon the lines that obtain in a fire station; each man has his specific duty, and after performing it looks to see that there is nothing more for him to do. Only a few seconds elapse between starting the first pump and starting the first turbine.

The turbine throttle is opened as fast as an 8-inch steam valve can be opened without endangering the steam piping system. It is not considered advisable to open the throttle valve as fast as a man's strength will permit; but if nothing unusual occurs in the pipe line, sentiment does not spare the turbine. One electrician attends to the switchboard and telephone. As soon as the machine approaches speed, the synchronizing system is cut in and the main switches are got ready. One and one-half minutes will do all the work here outlined, including the time taken in mustering the crew from various parts of the building, itself not a trivial matter.

Manipulating an engine regulator so that it shall be at a precise speed and at an exact phase relationship from some other machine, not more than 1-1500 part of a second removed from it, is no matter that can be hurried, and one minute is fast time on such work. But the whole thing, phasing-in and all, has been done in 2½ minutes, including full load on the turbine, which started from a standstill. This performance has been gone through a great many times, and our record book shows that out of 43 such calls, 10 starts were made in 2½ minutes, 18 in 3 minutes and 15 in 3½ minutes.

We have taken the time in a number of instances when all the auxiliaries have been in motion and it only remained to start the turbine and phase it in on the line; the only valves to open in such cases are the throttle and one small oil valve. The two quickest starts have been made in forty-five seconds and seventy seconds, respectively, including phasing in. Others range between one minute ten seconds and one and one-half minutes. These two quickest starts were made on a turbine which had stood for twenty-four hours with the throttle valve shut tight though there was a slight leakage past the seat. After the throttle valve is off its seat it is not more than thirty seconds before the turbine is up to speed. A cross compound reciprocating engine of the four-valve type, 2,000 horse-power capacity, can be brought up to speed from a standstill in five minutes if it is hot all over. This five minutes is to be compared with the seventy seconds required for the similar turbine operation.

A reciprocating engine, which is turning over slowly with the throttle valve just off its seat or with by-pass open and having all its oil cups open and regulated, can be brought up to speed, say seventy-five turns, in two and one-half minutes. This can be compared with the thirty seconds necessary for bringing the turbine up under the same conditions; that is, about one-fifth the time necessary for bringing up the engine. If the engine is cold all over and has all its oil cups shut tight, all its auxiliaries quiet, fifteen minutes is called a rapid start. Starts have been made under such conditions in twelve minutes. When we start a cold turbine, we open up the valve and let her turn, and in two minutes we are ready to bring her up to speed and she will be at speed in two and one-half minutes, dividing the engine's time by more than four.

SCRANTON MEETING OF THE A. S. M. E.

The annual meeting of the American Society of Mechanical Engineers was held at Scranton, Pa., on June 6 to 9 inclusive. The opening session was in the evening at the Y. M. C. A. hall, where usual courtesies were extended in the way of an address by a representative of the city and a reply by the president of the society. Following these preliminaries was the presentation of the papers assigned to the first meeting.

"The Transfer of Heat at High Temperature" by Prof. Frank C. Wagner, of the Rose Polytechnic Institute, summarized experiments to determine the time required to raise plates of iron and steel to a welding temperature in an open hearth regenerative furnace. This paper was of more interest to metallurgists and physicists than to mechanical engineers and is not suitable for abstracting here. A similar remark can be made in regard to several of the other papers presented at the several meetings, and in this report only those papers will be mentioned hereafter that are likely to interest MACHINERY readers, since a complete list of the papers was given in the last number.

"The Standard Unit of Refrigeration" by F. E. Matthews, of the De La Vergne Refrigerating Machine Co., New York, was a paper in which the author attempts to crystallize ideas on this subject, with the hope that it may lead to the final adoption of some standard unit of refrigeration.

The final paper of this session was upon "Some Types of Centrifugal Pumps," by William O. Webber, consulting engineer of Boston, Mass. This paper called out the most discussion of any, and is abstracted in the Engineering Review. It was pointed out in the discussion that centrifugal pumps might be used to advantage in oil pipe lines, and for many other purposes where plunger pumps are now used, although doubt was expressed as to its adaptability to water works duty unless the steam turbine should be so far developed that the two types of rotary machines, the turbine and rotary pump, when connected should be found suitable for this class of work.

Wednesday Morning Session.

This session was opened by a short business meeting, during which Mr. C. W. Hunt reported the progress of the plans for the new Carnegie engineering building. It was said that the architects and the plans had been selected with the greatest care, and it was believed that the plans would meet every requirement. They not only provide a building sufficient to house the three great engineering societies, but also provide sufficient accommodations to enable the building to serve as headquarters for other engineering societies, thus making it the great center of the country for men engaged in different branches of engineering work.

The question of the place for the next spring meeting came up and it was decided to accept an invitation from the engineers of Chattanooga, Tenn. It was represented that the city has many attractions for visitors and that its diversified industries are of unusual interest to those mechanically inclined.

Following the business meeting was a paper upon "The Microstructure and Frictional Characteristics in Bearing Metals," by Melvin Price. The investigation recorded in this paper was conducted at Columbia University, and was an attempt to trace the relation between friction and the composition and structure of bearing metals, although unfortunately the results obtained did not seem to have repaid the immense amount of labor required for the experiments.

Mr. W. J. Keep, Detroit, Mich., added to the large fund of information that he has contributed at various times upon cast iron and its properties, by a paper upon "The Crushing Loads and Microstructure of Cast Iron." This series of tests is valuable on account of its completeness and because a chemical analysis was made of each size of test bar used.

The most interesting paper of the meeting was by Professor Charles H. Benjamin, of the Case School of Applied Science, of Cleveland, O., upon "Smoke and Its Abatement." This paper is reviewed in the engineering edition of this month.

Wednesday Evening Session.

An interesting and practical paper upon "Steam Turbine Operation" was given by A. S. Mann, Schenectady, N. Y. He outlined provision made in a station containing Curtis steam turbines for placing the turbines in operation in an exceedingly short space of time when required by an emergency. The turbines serve as auxiliary power in a plant normally provided by water power and in case of the latter failing, turbines could be started in from two to three minutes, carrying full load. This paper appears in the engineering edition.

Mr. A. Bement, the specialist in combustion and boiler practice, of Chicago, presented two papers, one upon "The Efficiency of Steam Generating Apparatus," of which he had made special study, and the other upon "The Performance of a Superheater."

A new type of valve gear actuated by steam was described by W. A. Collier, Jackson, Tenn. The valves are of the Corliss type, operated by small steam cylinders to which steam is admitted and released by small valves controlled by a valve gear receiving motion from the engine shaft. This gear is said to give an unusually good steam distribution, to be cheap to construct, and after six months' use at a speed of 100 revolutions per minute, has shown no appreciable wear.

Thursday Morning Session.

This was the concluding professional meeting and was opened by a brief paper, given in our engineering edition, upon "Proportions and Heads of Machine Screws." It was shown that dimensions vary greatly among different manufacturers and a table of dimensions that was believed to be well adapted for general use was given.

Professor W. W. Bird, of the Worcester Polytechnic Institute, presented a new treatment of the already rather worn-out subject of "Belt Creep." The paper described the 4-inch and 6-inch single and double endless belts, and it was concluded that the belt creep should not exceed 1 per cent.

Another well-worn subject was introduced by Professor Chas. E. Lucke, upon "Laboratory Courses in Engineering Schools." He gave to the subject, however, a freshness of interpretation and treated it in a common-sense way which made it of unusual interest and won the approval of several educators who commented upon it. He criticised the usual method of instruction and held that the chief functions of laboratory work are first to teach the student how to make the standard commercial tests; second, to judge and criticise data and records in books and professional papers; third, to judge the limits of practical performance for machinery of all classes; fourth, to learn how to approach a new problem and to outline a method of attack preparatory to its solution.

The concluding paper was by Rear-Admiral Melville, upon "Epochs in Marine Engineering." This paper is the same that was presented at one of the local winter meetings of the society in New York, and has already been abstracted in MACHINERY.

As usual at the June meetings of the Society, the excursions were perhaps the most appreciated as well as the most useful part of the proceedings. At Scranton, the most prominent point of interest is the International Correspondent Schools, whose business has reached a magnitude that is probably not appreciated even by many technical men. This institution sends out as much mail as is dispatched by the whole of the city of Scranton, numbering 110,000 inhabitants. Demonstrations of the phonographic system of instruction in language were given. Inspection was made of the text books and other material published for the various departments, and the printing department elicited much favorable comment. This is a model plant, said to be one of the largest in the country. Visits were also made to the Allis-Chalmers Works, formerly the Dickson Machine Co., to the Lackawanna car shops recently completed, the Hampton Power Plant, equipped with Curtis turbines, and the Hampton water hoist, in which two buckets, each with a capacity of 17 tons of water, are operated electrically by a 800 H. P. motor, raising water from a depth of 500 feet. The complete cycle of one bucket is made in about one minute and 55 seconds.

On Friday, the last day of the meeting, a trip was made over the new third-rail road running between Scranton and Wilkesbarre. This line is familiarly known as the Laurel Hill line, is double-tracked, and is about twenty miles long. It owns its right of way and has a heavy traffic. The road is of unusual interest and is one of the best examples of electric rail-

way equipment in the country. The power plant is at Scranton, equipped with Westinghouse vertical engines operating generators in multiple, and supplying both direct and alternating current. There is a sub-station at a mid-way point on the line which converts the alternating to the direct current and reduces its voltage. A number of points of interest were visited at Wilkesbarre.

* * *

NEW LIFTING MAGNET.

A new lifting magnet has been brought out by the Electric Controller and Supply Co., Cleveland, O., of much larger capacity than the magnet illustrated in our May number. This magnet is shown in three views herewith and is of a type designed for handling stock such as pig iron, light or melting stock, small castings, bolts, rivets, iron turnings, etc. This magnet has a lifting capacity of 1,200 pounds of pig iron, small castings, or of bolts, nuts, etc. It will lift 1,100 pounds of medium-weight castings, and a "skull-cracker ball," used for breaking up scrap in the foundry yard, of 11,000 pounds

Fig. 1. Electro-magnet Lifting Machine-cast Pig Iron.

weight. It will be seen from this that a lifting magnet will raise a much greater weight when the latter is one piece than when there are a number of small pieces which not only cling to the magnet, but to each other.

A magnet of the type shown is adapted for loading or unloading cars, piling stock, and for general carrying purposes. Assuming its average lift to be 1,000 pounds, and the work to consist of loading from cars into furnace-charging boxes for the foundry cupola: if the crane man can make a transfer in 45 seconds, or at the rate of 90 per hour, and the crane works eleven hours in each twelve-hour turn, a crane using this magnet will have a loading capacity of $1,000 \times 90 \times 11$, divided by 2,000, or 494 tons per turn. With this magnet, therefore, a 5-ton crane could serve four such furnaces with ample safety, thus increasing the capacity of the crane besides saving the labor ordinarily used for filling the boxes. For breaking scrap, the magnet takes the place of the ordinary latch hook used for raising the skull-cracker and will lift the bolt in any position, making it unnecessary to waste time in getting the bolt so located that it can be attached to the hook of the hoisting rope. There is no necessity for any yard men to be near when the ball is dropped, as the magnet is controlled by a hoist operator at a safe distance and there is therefore no danger from flying scrap. Another advantage of the magnet for this work is that there is no jerking of the supporting rope as is the case when the latch is released, with the ordinary apparatus, so that the aim is sure and every blow effective.

Fig. 2. Mass of Iron Turnings Lifted, showing extent of Magnetic Field.

In the accompanying illustrations, Fig. 1 shows one of these magnets lifting machine-cast pigs from a deep pile and gives an idea of the rapidity with which this class of material can be moved either in the yard or in loading or unloading cars without having to handle it by means of manual labor. To raise his load the crane operator has simply to turn on the current and set the crane in motion; while to deposit the load he lowers the magnet and turns off the current. Fig. 2 shows one of these magnets that has picked up a load of iron turnings from a pile, and the shape of the mass of turnings that has been raised indicates the extent of the magnetic field of the magnet.

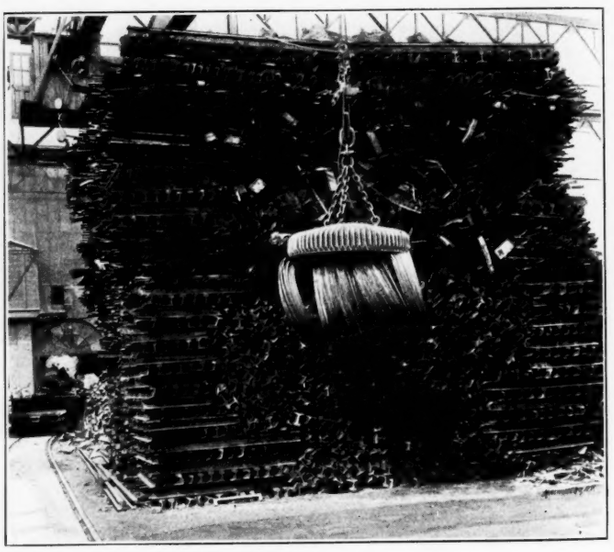


Fig. 3. Magnet Carrying Wire Coils.

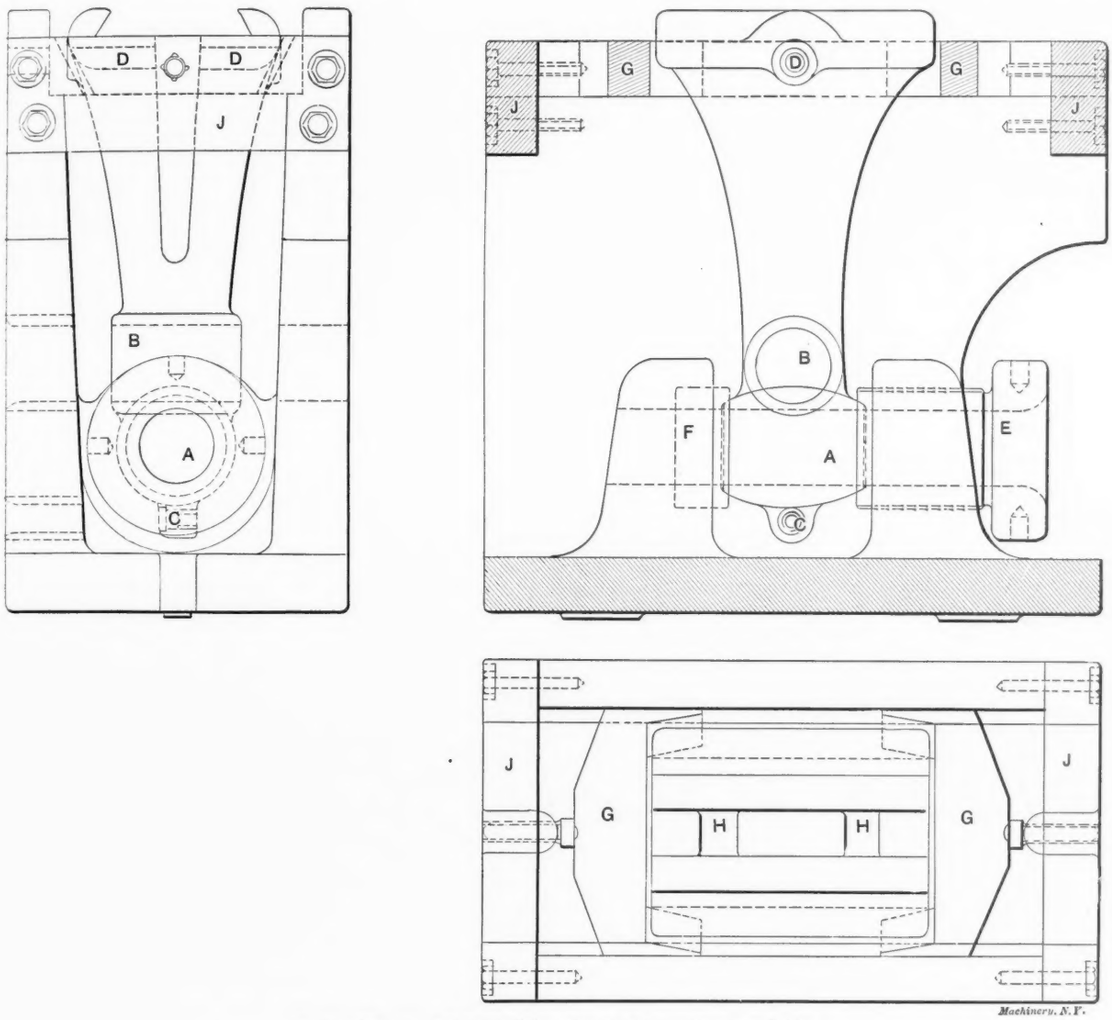
Fig. 3 shows a magnet carrying several coils of wire, the magnet being suspended from the crane, one end of which appears in the upper left-hand corner of the engraving. The pile of scrap at the rear was piled by the magnet, and it will be noted that it extends up almost to the level of the crane girders. By the use of magnets it is possible to handle stock to a greater height than in any other way, because less head room is required for attaching to the crane hook.

LETTERS UPON PRACTICAL SUBJECTS.

MILLING AND DRILLING JIG.

Editor MACHINERY:

I send you three views of a jig for milling and drilling the sliding heads of small sensitive drill presses. It is another example of the type of screw bushing jigs described in the December, 1904, issue, but with some slight modifications. One of the essential requirements of the jig was that the hole, *A*, which receives the spindle, should be accurately bored in the center of the boss so that the machine when finished should present a pleasing appearance to the eye. It was also considered necessary that the dovetailed groove should be milled so as to make the heads interchangeable.



Jig for Milling and Drilling Sliding Heads of Small Sensitive Drills.

Previous to making this jig the castings had been planed a certain distance from the center of the cored hole *A* and this hole had been afterward bored and reamed in a jig which received the planed surfaces. By this method the hole *A* was quite often out of center with the outside of the casting because it was very seldom that the core was placed in its proper position in the mold to insure its being central. To obtain somewhere near satisfactory results it was necessary that the planing should be carefully done and that the sides of the dovetail be equally distant from the center of the hole. Hole *B* for the pinion shaft which feeds the spindle, holes *D D* for the clamping screw and lever, and hole *C* to adjust the tension of the feed sleeve had been laid out by hand and drilled after planing. The excessive cost of finishing the heads by this primitive method had long been a source of worry to the firm and the jig was therefore built with the understanding that it should show a noticeable reduction over the time required by the previous method. It was hardly expected to make a very large saving because the jig and casting being heavy were not as easily handled as the casting alone and, added to this, was the necessity of keeping both the vertical miller and the drill press tied up on this job. Laboring under these difficulties it was gratifying to note that the first lot came through with an average saving of ten minutes on each casting and also the additional gain in the improved appearance and interchangeability of each head.

The body of the jig, made of cast iron, is finished all over its base, slotted and tongued to fit the milling machine table and cut away at one side to allow for tightening the screw bushing, *E*. The casting was located and held in its proper position in the jig by means of the large screw bushing *E* clamping against the stationary plain bushing *F*. To take the thrust of the milling cut the sliding clamps *G G* were tightened evenly against both ends of the casting by the square head setscrews shown. These setscrews were fitted to the two steel straps *J J* which strengthen the jig across the top where otherwise it would be open and weak to resist the action of the clamps. The peculiar shape of these clamps should be noted; they slide along and are seated in a groove milled the entire length of the jig, and it is obvious that when they are brought up in place no movement of the casting in the jig can occur.

After the casting has been thus properly secured in the jig it is placed on the milling machine and roughed out with a cutter of the correct shape. It is then finished level with the top of the steel straps which are used as a gage for setting

the depth of finishing cut on the first casting. The micrometer adjustment on the elevating screw is used for the following castings.

The well-known difficulty of accurately drilling and reaming cored holes has been overcome in this case by the use of a roughing and finishing single pointed boring tool on the holes *A* and *B* before putting through the machine reamer, the finishing boring tool leaving about 0.005 inch for the reamer to take out. It is of course unnecessary to add that the reamer and shank of the boring tool should be a good fit in their respective bushings and likewise that the screw bushing should be a good fit in the jig. The hole *c* having a counter-bored shoulder for a filister head screw is finished before removing the casting from the jig in the following manner: The bushing in the jig is made the size of the head of the screw and a drill of this size is used to spot the hole. It is followed by tap and clearance drill and lastly by a counter bore, which is nothing more than a piece of drill rod of the proper size having teeth cut on the end.

An improved method of fastening together parts of jigs and fixtures is also shown. The steel straps *J J* are fastened to the cast iron body of the jig with hexagon head cap screws the heads of which are sunk below the surface and tightened up with a socket wrench. This necessitates a counterbored hole the size of the outside of the socket wrench but has the advantage over the filister head screw of being able to be set up as tight as a cap screw with a protruding head.

New York. H. J. BACHMANN.

AN UNCOMMON LATHE JOB.

Editor MACHINERY:

A peculiar job came under my notice some time ago. The top portion of a rudder stem had worn very badly and it was decided to have it repaired. The original diameter of the head was about 6 inches, and it had worn down in the socket under the tiller to about 3½ inches. The stem was longer than necessary and instead of cutting the defective part off

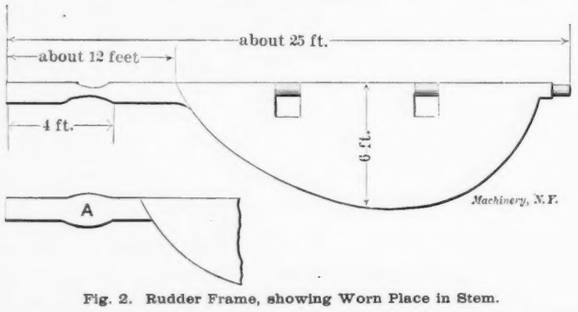


Fig. 2. Rudder Frame, showing Worn Place in Stem.

and welding a new end on, the end was staved up by the blacksmiths until there was sufficient excess metal at the worn part to turn down to the original diameter. The question was how to machine the part that worked in the gland. The sketch, Fig. 2, will better explain the difficulty. The rudder frame was in one piece and had to remain intact. No lathes were available that could swing the rudder between its centers, and the part marked *A* had to be machined fairly true with the main axis of the rudder. The following plan was adopted and proved successful.

The longest lathe available, 28 inches swing and about 18 feet in the bed, was requisitioned, the tailstock removed and all the top gear taken off, the main saddle carrying the tool rest. The rudder was then fastened to the saddle, being lined up as near as possible so that the axis of the rudder would coincide with the line between head and tail centers of the lathe under ordinary conditions. The overhanging portion of

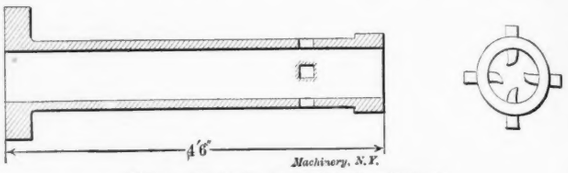


Fig. 3. Cast Iron Pipe, Carrying Cutters.

the rudder was supported by rollers on trestles. A cast-iron pipe as per sketch, Fig. 3, was fixed in the chuck, the flanged end being fastened to the chuck and the tail end trued up on the outside to allow it to run in a V-guide or steady rest. Short cutters were inserted in the four square holes with the cutting edges inside. The saddle was then moved forward, partly carrying and partly dragging the rudder. The feed gear

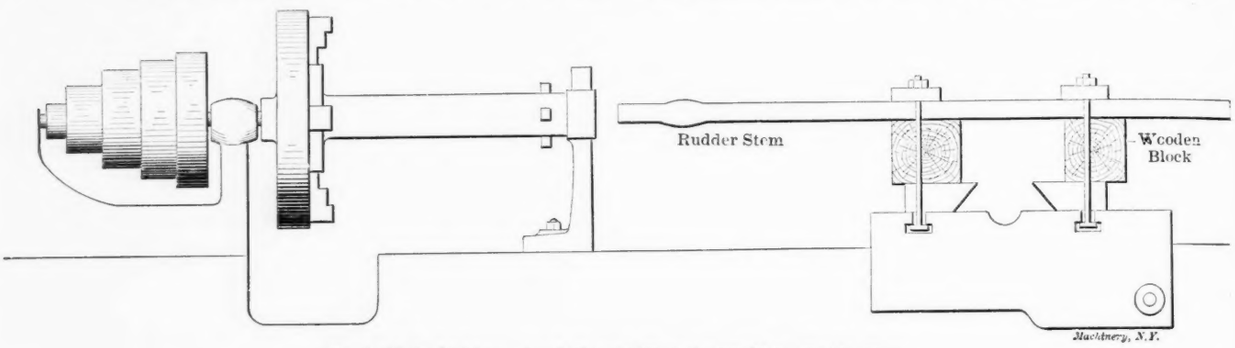


Fig. 1. Method of Arranging Lathe and Work, for Machining Rudder Stem.

was then set in motion and the swelled end of rudder was steadily carried on through the pipe, the cutters doing their work in the one setting. STRAIGHTEDGE.

COVERING PULLEYS WITH LEATHER.

Editor MACHINERY:

I recently saw a belt, which though tightly strained, slipped badly because the contact with the smooth pulley rim was inadequate to enable the belt to drive the work required of it. The poor performance of duty rendered by this belt and pulley caused me to wonder why such a condition often occurs when it can so easily be corrected by a method that insures the maximum of adhesiveness, i. e., covering the pulley with leather or rubber. Haswell's Engineer's and Mechanic's Pocket Book, 22d edition, states as results of experiments upon the adhesion of india rubber and leather belting by I. M. Cheever that a rubber belt slipped on iron pulleys at 90 pounds; on a leather or leather-covered pulley at 128 pounds; and on a rubber or rubber-covered pulley at 183 pounds. Also that a leather belt slipped on an iron pulley at 48 pounds; on a leather or leather-covered pulley at 64 pounds; and on a rubber or rubber-covered pulley at 128 pounds. It can be seen that the leather-covered pulley greatly exceeds iron in the adhesion of the belt, and rubber gives more adhesion than any other material used on the periphery of pulleys.

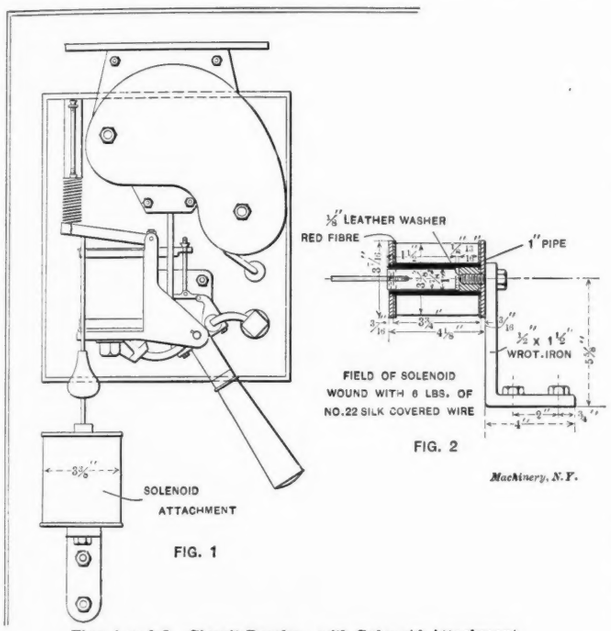
I have never used rubber-covered pulleys, but have seen pulleys on which the belts slipped badly, which, when covered with leather, showed such an improvement in the adhesion of the belt that all slipping of the belts stopped. The method of attaching the leather covering in this case was performed as follows: A leather belt the exact width of pulley was fastened to the pulley face by a pair of hand screw clamps placed within one inch of the end of the belt. The belt was stretched and wound around the periphery of the pulley tightly, and the ends lapped. The joint was marked with a sharp knife and the belt was then removed. The pulley face was painted with a coat of good white lead ground in oil, and when the paint had dried somewhat, but was still a little sticky, the face of the pulley and the grained surface of the belt were covered

with a coat of good glue. One end of the belt was then fastened to the pulley face and secured by a screw clamp as before. Then as the belt was tightly wound to its place on the face of the pulley, clamps were tightened as close as they could be applied and screwed up tight. The ends of the belt were brought together and the joint covered with a clamp which was screwed tight. The clamps were left on till the glue was dry and when they were removed the face of the pulley was neatly covered with a leather jacket on which the belt never slipped. The adhesion of the two leather surfaces transmitted the motion of the entire circumference of the pulley at each revolution with nearly the same certainty as gear teeth.

In my own individual experience a man owning a saw mill wished me to prevent his belt from slipping, which it did badly, when the saw was buried deep in a log. The pulley was 24 inches diameter and the saw 60 inches. As the mill was damp and the job in haste, I did not stop to bother with paint and glue but secured the leather covering to the face of the pulley by drilling a row of holes for copper rivets around the periphery, on each edge of the pulley, ⅝ inch from the edge and 2 inches apart; also a row of holes around the center of the pulley 5 inches apart to hold the leather tight to the pulley in the center. Two rows were drilled across the face of the pulley to secure the ends of the leather. I placed the heads of the rivets outward, sinking them a little below the surface of the leather with a drift of the same diameter as the rivet heads, hammering them tight on the inside of the pulley rim. On this covering the belt never slipped even when the saw was entirely buried in a log so large that the teeth of the saw did not cut through the top of the log. I instructed the sawyer to tighten the rivets occasionally, and by this simple job a mill that had given much trouble became a source of comfort and profit to the owner and the men who did the work.

C. E. MINK.

Syracuse, N. Y.



Figs. 1 and 2. Circuit Breaker, with Solenoid Attachment.

CIRCUIT BREAKER, WITH SOLENOID ATTACHMENT.

Editor MACHINERY:

The cut shows a design of circuit-breaker with a solenoid attachment. These circuit-breakers are of a well-known type (General Electric) and are on the circuit of a 100 H. P. motor which drives a lineshaft and to which are belted three large rock crushers. These crushers are at some distance from the switchboard, and as it often happens that the crushers must be shut down quickly, considerable time would be lost in running to the switchboard and throwing the switch. By using this solenoid attachment the motor can be stopped from each crusher by simply throwing a switch.

The solenoid, Fig. 2, is operated by a 250-volt current. When the knives of the switch are thrown together the armature is drawn in and operates the circuit-breakers. After the motor has been stopped the attendant can then throw the switch on the switchboard.

Desloge, Mo. HERBERT S. GLADFELTER.

RULES FOR FIGURING THE SPEED OF PULLEYS AND GEARS.

Editor MACHINERY:

A few days ago a machinist of my acquaintance came to me for the purpose of ascertaining the method of finding the proper size of pulley to use to drive at a certain speed a machine which he was erecting. I was a little surprised at his lack of knowledge in the matter but upon second thought I have found from time to time in my experience among mechanics that the rule which seems to me to be very simple is not so generally known as one would suppose. This table will show the notation used in the formulæ:

a = Diameter of driver.
b = Diameter of driven.
c = R. P. M. of driver.
d = R. P. M. of driven.
R. P. M. = Revolutions per minute.

The diameter of driver, R. P. M. of driver, and diameter of driven being given, to find the R. P. M. of driven, then

$$\frac{a \times c}{b} = d$$

The diameter of driver, R. P. M. of driver, and R. P. M. of driven being given, to find the diameter of driven, then

$$\frac{a \times c}{d} = b$$

The diameter of driven, R. P. M. of driven, and diameter of driver being given, to find the R. P. M. of driver, then

$$\frac{b \times d}{a} = c$$

The diameter of driven, R. P. M. of driven, and R. P. M. of driver being given, to find the diameter of driver, then

$$\frac{b \times d}{c} = a$$

To make the rules perfectly clear to apprentices and all others to whom the subject may be new, let us substitute figures in the first formula. Suppose, for example, we have a driving pulley (or driver as termed in the rules) 24 inches diameter, making 300 revolutions per minute, and the driven pulley is 16 inches diameter, then to find the revolution per minute of the driven pulley we have: $\frac{24 \times 300}{16} = 450$ revolutions per minute of driven pulley.

The fundamental principle of the rules, of course, is to multiply the diameter of the pulley by its own revolutions per minute and divide by the other given term to obtain the result, and the same rules may be used in determining the speed of gears by substituting the number of teeth for the diameter.

FREDERIC E. DURFEE.

Fall River, Mass.

TO FILE A SHARP SCRIBER.

Editor MACHINERY:

To file a scriber point take ordinary drill rod of any size desired and catch same in a lathe chuck, leaving say about ⅞ or 1 inch projecting. Then with a good sharp smooth file start the point by filing it blunt say at about an angle of 60 degrees, then gradually file the angle sharper and sharper by merely changing the position of the file (which should be at least 1 inch wide) so it stands more and more in a line parallel with the piece being filed. By this method it is possible to get a long slim point, whereas if the file is held at the finished angle to begin with, difficulty arises due to the fact that the file has too much surface to cut and the point gets weak and springy. Finish point by holding on side of good straight emery wheel or disk grinder (while soft) so the grinder marks run from point to back; after hardening, oil stone only.

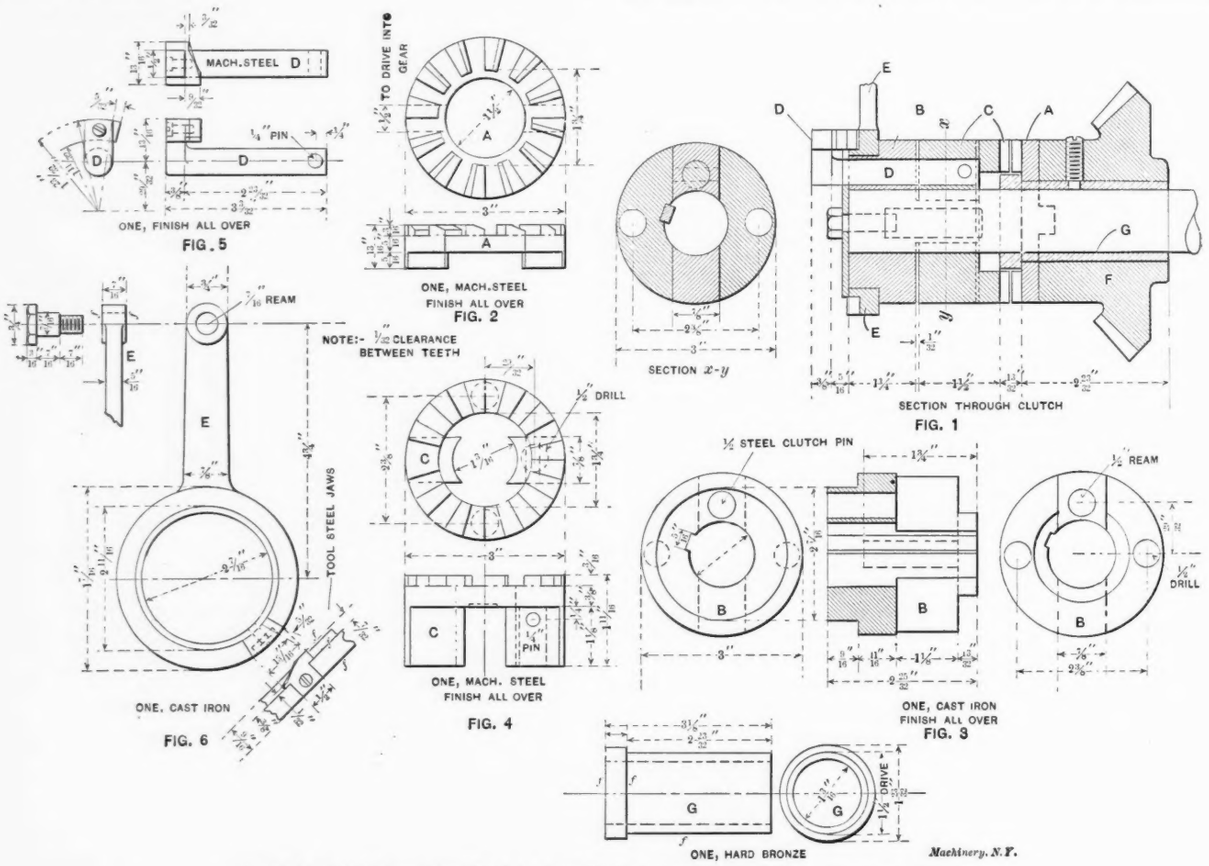
Chicago, Ill. R. A. LACHMANN.

ONE-TURN CLUTCH.

John D. Adams.

Editor MACHINERY:

Very often in the design of special machinery, we have certain operations, such as feeding, etc., which are done by hand and which require that certain parts remain at rest during the period. Sometimes this period of rest can be taken care of by intermittent gears or similar device, but when the period of rest is of uncertain duration these are very unsatisfactory. The drawings herewith show a clutch so constructed that by means of a slight movement of a lever it will make exactly one turn (or one-half or quarter turn if so designed) and then stop exactly at the point where it started.



Clutch by which Intermittent Shaft is given Motion of exactly one Turn or Fraction of Turn.

Fig. 1 is a cross section lengthwise of the shaft which has the intermittent motion. The bevel gear, *F*, mounted on the bronze bushing *G*, furnishes the power and runs continually. The circular ring, *A*, Fig. 2, has twelve teeth and is held up against the end of the gear by the brass bushing, *G*. The end of the gear is slotted radially to receive the two lugs on the back of Fig. 2, which cause it to turn with the gear. The piece *B*, shown in detail in Fig. 3, is keyed to the intermittent shaft, and has two slots ⅞ inch wide diametrically opposite to each other. These two slots are to keep the set of teeth shown in *C*, Fig. 4, from turning on the shaft and at the same time to allow them to move forward so as to engage the set of teeth on the end of the bevel gear. This forward motion of Fig. 4 is controlled by the clutch pin, *D*, shown in Fig. 5, also clearly shown in the section view, Fig. 1. This clutch pin passes freely through the ½-inch hole in Fig. 3 and is secured by a ¼-inch pin to one of the lugs on the back of the intermittent set of teeth, *C*, Fig. 4.

The clutch is operated by the lever *E* shown in Fig. 6, which has a slight motion around the end of the block, Fig. 3. When the clutch is at rest the highest point of the inclined plane on the clutch pin, *D*, Fig. 5, rests on the highest point of a similar plane on the controlling lever, Fig. 6. Two coil springs placed in diametrically opposite holes in Fig. 3, press against the back of the intermittent set of teeth, *C*, and keep the two inclined planes just mentioned continually bearing against one another so that when the lever, Fig. 6, is slightly rotated the clutch pin drops off allowing *C* to engage clutch *A*, which rotates the shaft. A spring, not shown, brings the controlling lever back to its original position as soon as the momentary pressure of the hand is released and when the clutch pin, which travels around with the clutch, nearly completes one turn, its inclined plane rides up on the one attached to the controlling lever and the two sets of clutch teeth are pulled apart, thus stopping the motion of the intermittent shaft. At any convenient point around the mechanism to be operated a definite stopping point should be arranged and so connected with the controlling lever that the stopping point and the lever will be moved simultaneously. In this way it is possible to make the clutch stop at precisely the same point every turn. By placing two, three or four inclined planes on the controlling lever the clutch may be made to stop a corresponding number of times each turn.

Phœnix, Ariz. JOHN D. ADAMS.

MR. JOHN D. ADAMS was born in Toronto, Canada. His technical education was acquired in the Toronto Technical School. He at present holds a position with the Department of Interior at Phœnix, Arizona, and prior to this engagement was with Park, Davis & Co., Detroit, Mich. His specialty is automatic machinery.

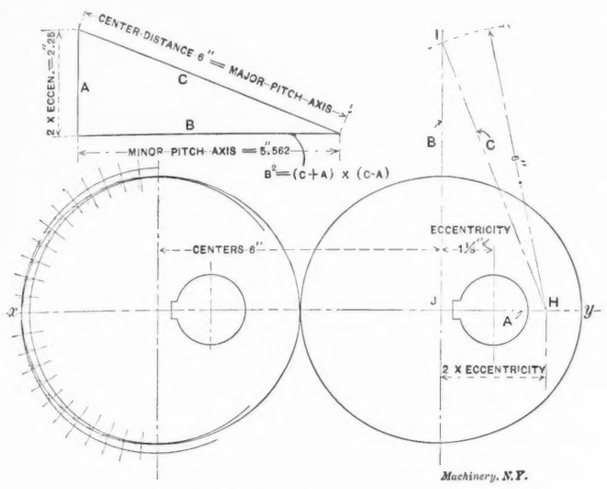
LAYOUT OF PITCH CIRCLES OF ELLIPTIC GEARS.

Editor MACHINERY:

A short time ago it was my pleasure to draw a pair of elliptic gears, and, as very little seems to have been written regarding this style of gearing, perhaps my endeavors may be appreciated. The sketch herewith shows the method employed for laying them out.

The center distance of gears was 6 inches and the eccen-

tricity 1⅛ inches. A horizontal line $x\ y$ is drawn, which represents a center line and the major axes of both gears. Two perpendicular lines which represent the minor axes of the gears are then drawn 6 inches apart; this distance also equals the major pitch diameter, which can be readily spaced off. The minor pitch diameter is obtained graphically by placing a point H a distance of double the eccentricity from the minor axes on the center line, and from this point with a radius equal to the center distance, draw an arc cutting the perpendicular line at I. The distance from I to J equals the minor pitch diameter.



Layout of Pitch Circles of Elliptic Gears.

From this we see that the major pitch diameter C = center distance of gears; minor pitch diameter $B = \sqrt{(C+A)\ (C-A)}$ and eccentricity $E = \sqrt{R^2 - R_1^2}$ where A = double the eccentricity, C = center distance, R = half of major pitch diameter, and R_1 = half of minor pitch diameter.

After obtaining the diameters the ellipse can be drawn by any of the numerous methods. Perhaps the most accurate for any diameters is to point off equal (or unequal) spaces on a circle whose diameter equals the major axis, and through

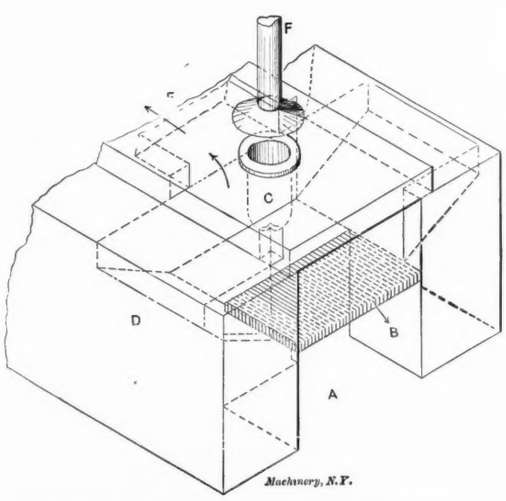


Fig. 1. Coke or Charcoal Furnace for Heating Bath.

these points draw short radiating lines. From the same center draw another circle whose diameter equals the minor axis. Through the points in the larger circle erect short vertical lines, and where the radiating lines intersect the smaller circle draw horizontal lines.

The intersection of these lines will be points of the ellipse, which can be sketched in free hand or by aid of the compass.

Bridgeport, Conn. C. E. JOSSELYN.

HEATING BATHS FOR USE IN STEEL HARDENING.

Editor MACHINERY:

Where it is desired to heat small pieces in melted lead, cyanide of potash or other substance which enables a foreordained even temperature to be attained, and which at the same time protects the articles which are to be heated from the oxidizing action of the air more conveniently than is possible by the use of muffles, the furnaces shown in the two annexed illustrations will serve as types.

In Fig. 1 the crucible C is to be heated by mild coke or still better by charcoal, as this gives a more regular heat. The grate must be adapted to the fuel. Where coke is used, a brand should be employed which is known as "extra mild"; otherwise despite the regular nature of the coke fire there might be trouble with overheating, which would entail loss of time in allowing the melted material to cool down to the desired maximum temperature.

The furnace shown in Fig. 2 is intended for either wood, anthracite or soft coal. The temperature of the bath between such points as represent fusion of the solid and evaporation of the liquid is extremely difficult to estimate from the general appearance of the latter, particularly as the apparent color depends on the light of the room in which the furnace stands, on whether the observer's eye is accurate or not, and on whether he has already made several such observations—in which latter case he will be very apt to estimate too low. A good pyrometer will do "yeoman's service" in preventing expensive mistakes.

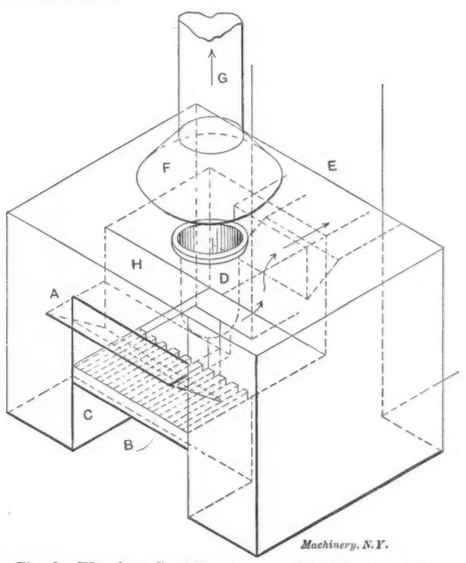


Fig. 2 Wood or Coal Furnace used for Heating Bath.

The fumes of the lead are injurious to the health, and those of the cyanide are highly poisonous; therefore, both these must be conducted away by a special inverted funnel, F, Fig. 1; G, Fig. 2, that has a strong natural or artificial draft entirely independent of that of the smoke passage, E, both figures.

In Fig. 2 the firing is done through a hopper, A. The ashpit, C, can be closed by a plate so as to lessen the draft once the fire on the grate B has caught.

Hanover, Germany. ROBERT GRIMSHAW.

TABLES GIVING ANGLES OF THREADING TOOLS.

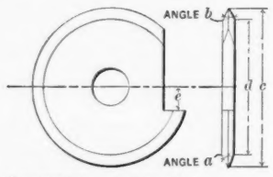
Editor MACHINERY:

Herewith I contribute three sets of tables which may be of interest to some of your readers. The first table in question deals with the circular threading tool. This kind of a tool most generally has its cutting face below its center line, which, of course, changes the angles; it not only changes the angles, but as we lower the cutting edge, the same becomes a convexed line. On large diameters this second error is not noticeable to any extent, although it exists from the very moment we lower the cutting face of such a tool. This

one item makes it very difficult to accurately give the reader the angle to make such a tool, which should cut an accurate 60 degree thread, when the tool is cut a certain amount under its center line.

The angles given in Table No. 1 were computed by the writer taking diameter d to be 1/8 inch smaller than c; this difference was used throughout the tables. The first column gives the largest diameter of tool, while the second gives the diameter from which the following angles were obtained. The eight columns following give the angle of one side of tool and also the included angle for such tools which have their cutting face 1/8, 3-16, 1/4 and 5-16 inch below the center line.

TABLE NO. 1. CIRCULAR THREADING TOOLS.



a = ANGLE WITH SIDE OF TOOL
b = INCLUDED ANGLE OR 2 X ANGLE a
c = LARGEST DIAM. OF TOOL
d = c − .125 (THESE TABLES WERE COMPUTED FROM THIS RATIO)
e = DISTANCE FROM CENTRE LINE TO CUTTING FACE

LARGEST DIAM. OF TOOL = c	DIAM. d = c − .125	ANGLES ON CENTRE LINE WHEN e = .125		ANGLES ON CENTRE LINE WHEN e = .1875		ANGLES ON CENTRE LINE WHEN e = .250		ANGLES ON CENTRE LINE WHEN e = .3125	
		ANGLE a	ANGLE b	ANGLE a	ANGLE b	ANGLE a	ANGLE b	ANGLE a	ANGLE b
3/4	5/8	31° 41′	63° 22′	34° 38′	69° 08′	40° 10′	80° 20′		
7/8	3/4	31 15	62 30	33 05	66 10	36 18	72 36	42° 25′	84° 50′
1	7/8	30 56	61 52	32 13	64 26	34 20	68 40	37 50	75 40
1 1/8	1	30 43	61 26	31 41	63 22	33 13	66 26	35 34	71 08
1 1/4	1 1/8	30 34	61 08	31 19	62 38	32 29	64 58	34 12	68 24
1 3/8	1 1/4	30 28	60 56	31 04	62 08	31 59	63 58	33 18	66 36
1 1/2	1 3/8	30 23	60 46	30 53	61 46	31 37	63 14	32 40	65 20
1 5/8	1 1/2	30 19	60 38	30 45	61 30	31 22	62 44	32 13	64 26
1 3/4	1 5/8	30 17	60 34	30 38	61 16	31 09	62 18	31 52	63 44
1 7/8	1 3/4	30 14	60 28	30 32	61 04	31 00	62 00	31 36	63 12
2	1 7/8	30 13	60 26	30 29	60 58	30 52	61 44	31 23	62 46
2 1/8	2	30 11	60 22	30 25	60 50	30 45	61 30	31 13	62 26
2 1/4	2 1/8	30 10	60 20	30 22	60 44	30 40	61 20	31 04	62 08
2 3/8	2 1/4	30 09	60 18	30 20	60 40	30 35	61 10	30 57	61 54
2 1/2	2 3/8	30 08	60 16	30 18	60 36	30 32	61 04	30 51	61 42
2 5/8	2 1/2	30 07	60 14	30 16	60 32	30 29	60 58	30 46	61 32
2 3/4	2 5/8	30 06	60 12	30 14	60 28	30 26	60 52	30 42	61 24
2 7/8	2 3/4	30 06	60 12	30 13	60 26	30 24	60 48	30 38	61 16
3	2 7/8	30 05	60 10	30 12	60 24	30 22	60 44	30 34	61 08
3 1/8	3	30 05	60 10	30 11	60 22	30 20	60 40	30 32	61 04
3 1/4	3 1/8	30 04	60 08	30 10	60 20	30 18	60 36	30 29	60 58
3 3/8	3 1/4	30 04	60 08	30 09	60 18	30 17	60 34	30 27	60 54
3 1/2	3 3/8	30 04	60 08	30 09	60 18	30 16	60 32	30 25	60 50
3 5/8	3 1/2	30 03	60 06	30 08	60 16	30 15	60 30	30 23	60 46
3 3/4	3 5/8	30 03	60 06	30 08	60 16	30 14	60 28	30 22	60 44
3 7/8	3 3/4	30 03	60 06	30 07	60 14	30 13	60 26	30 20	60 40
4	3 7/8	30 03	60 06	30 07	60 14	30 12	60 24	30 19	60 38

Machinery, N. Y.

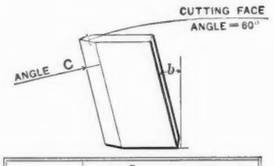
Should one desire to construct such a tool for very coarse threads, say, for instance 2 or 3 pitch, he can readily do so with very accurate results by simply manipulating the figures in the table. Example: A circular tool is to be made, of which the extreme diameter is to be 2 inches and which is to be used for cutting 2-pitch threads; its cutting face is to be 5-16 inch below the center line, and it must cut to the depth of 0.433 inch when the top width of cut equals 0.500 inch. Now the table gives us for a 2-inch diameter 60-degree tool cut 5-16 inch below center, the half angle 31° 23′, or 62° 46′ included angle. These angles would be accurate for making a tool that was to be used on threads that have an approximate depth of about 1-16 inch, but for the tool in question we would come nearer right if we reckoned our two diameters, namely, 2 inches for the one and 1 1/8 inches for second diameter, since $2 \times 0.433 = 0.866$, which is nearly 7/8 inch and 2 inches—7/8 inch would equal 1 1/8 inches for the second diameter. Now if we consider the intermediate diameter 1 9-16 inches—which is found as follows: $2 - 1\frac{1}{8} = \frac{7}{8}$ inch; $\frac{7}{8} \div 2 =$ 7-16 inch; 7-16 + 1 1/8 = 1 9-16 inches—we find upon referring to the tables that a 1 9-16-inch diameter is not given, so we divide the difference between a 1 1/2-inch and 1 5/8-inch diameter; this difference is 27 minutes, half of which would be about 14 minutes. This added to the angle given in the 1 5/8-inch line would equal 32° 27′, which would be the proper angle to make such tool.

We will now turn to the straight threading tool, which in one sense of the word is a more accurate tool than the circular variety, simply because we have not the convexed side to contend with. The cutting edge of a straight tool is always a straight line (provided it is made accurately) regardless of what the clearance angle is, although we have the same problem to contend with in this style of tool as in the circular variety, namely: when the cutting angle equals 60 degrees, for example, what is the angle on forward side of tool?

Table No. 2 gives this information. As will be seen in this table, the first column gives the clearance angles which range from 8 to 20 degrees, inclusive. In the second and third columns are the respective single and included angles, which when measured on the forward side of the tool will coincide with a perfect 60-degree angle on the cutting face.

There is still another item which is of no less importance than any previously mentioned, and which concerns both the straight and circular threading tools, and that is the setting of such tools in the machine so that they may stand in alignment with the angle of the thread that is being cut. Many threads are cut which are smooth on one side and rough on the other; the cause is not having an equal amount of clearance on each side of the threading tool. For instance, take any style of cutting tool which has two or more cutting teeth and clear the ones that work on one-half of the circle more than the other half, and the result will always be unsatisfactory. This is especially noticeable on turret forming tools of a screw machine. The old style of lathe tool which was used for threading purposes has a little advantage over the circular and straight tools in this item, because they have clearance both ways, but with such tools that can be ground without changing their form, we must obtain one clearance from the forward side only. This makes it more essential to have these tools stand as near in line with the angle of the thread as possible. But when we speak of setting such tools perfectly in alignment with the angle of a thread, we have an impossibility to contend with, because the root of a thread is always smaller in diameter than the apex and as the lead on both root and apex remains the same, the angle must of course change, when going from one diameter to another. In other words, the angle of the spiral at the root diameter is always greater than at the apex of thread. Now to compute our angle. I suppose the most correct diameter to select would be about midway

TABLE NO. 2. STRAIGHT THREADING TOOLS.



ANGLE b = CLEARANCE ANGLE OF TOOL	ANGLE C = ANGLE MEASURED ON FORWARD SIDE OF TOOL. C = ANGLE WITH CENTRE LINE		C = INCLUDED ANGLE	
8	30°	15′	60°	30′
9	30	18	60	36
10	30	23	60	46
11	30	28	60	56
12	30	33	61	06
13	30	39	61	18
14	30	45	61	30
15	30	52	61	44
16	30	59	61	58
17	31	07	62	14
18	31	16	62	32
19	31	25	62	50
20	31	34	63	08

between the root and apex of thread, but as the changes of angles are very slight, and really too slight to practically work to, the writer has constructed Table No. 3 which was computed from the diameter over the apex of the thread. This table fills the want very well as it gives the angles of all the threads that are in every day use on diameters which range from 1-16 inch to 6 inches. The accompanying formula can be used if one wants to figure out any other special pitch on diameter which is not contained in the table. When cutting double or triple threads all one needs to do is multiply the given angle by two or three, as the case may be. This table facilitates matters when making tools for cutting square

threads; it also comes in very handy when making tools for cutting spiral springs from the solid, which are so extensively used on sub-press die work. JOS. M. STABEL.

Rochester, N. Y.

"A BLUFF THAT WENT."

Editor MACHINERY:

Sam was the boss of the plating-room at the lock shop. It was a small shop, and it wasn't much of a job, but then Sam wasn't much of a plater, so he fitted in first-rate. He got eighteen straight per week, and that was good money down there. But one morning Sam went down to the shop to find the plant burned down and a month's pay due him, and no show of getting it until the shop started again. What to do in the meanwhile was the question. Now Sam was well acquainted with the house which had furnished the lock company with their plater's supplies, and he sent them a letter telling just how the case stood, and asked them if they could fling a job his way. An answer came back in three days with a special delivery stamp on the letter, telling him to go to The Country Hardware Company, that they had had a row with their plater, and he had "jacked up" and left them. They were in a fix and wanted a man bad.

The letter didn't say so, but Sam got the idea that the firm was not over liberal on the pay-roll end, but it was not a time for him to be "squeamish," so up he went. Well, it was a little backwoods place run by water-power. The shop was down in a hollow on the edge of a pond, and the boys could fish out the plating-room window when they got their vats full. The case was like this: They had a good man and he was a worker too, but the firm didn't know a good thing when they saw it, and they had pushed him harder than flesh and blood could stand, and one day he got on his ear and lit out. The firm said they thought they could save his wages by letting the oldest boy run the room; said it had worked fine for a while, but now the work was coming rough and it would peel off in spots.

The "super," who was also the owner, took Sam into the plating room at once, and showed him the work, and it did look bad. Sam looked the room over pretty well, and then put his hydrometer into the bath and tried his litmus paper to see if the solution was right. It was O. K. and Sam knew at once that the fault was in the potash. He never let on a word, but gravely dipping his hand in the solution and letting it drop off his fingers (a test by the way, that no man can tell a thing by) said: "Yes, the tubs are in a bad shape, and it will be a job to save them, but I can do it all right."

"Well—what wages do you want?" asked the "super."

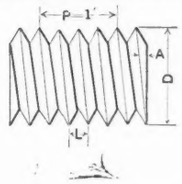
Now Sam had learned that the last man had run the room for about one-half of what it was worth, so he thought the concern should pay for all that they had coming this time, and he set his price at $24.00 per week. The "super" nearly fell off the stool—"What! $24.00?" said he. "Why we only paid our last man $12.00, and he didn't earn that; he used to sit round the room half the time and let the boys do the work."

"All right, if you want a cheap man, get him; the woods are full of them, but if you let those tubs run a week more, no man can save them, and you know what a new lot of acid would cost you. That solution is worth twenty cents a gallon, and you have about five thousand gallons there." That settled it; and Sam was hired at his own price.

He pulled off his coat and went to work. The old potash

TABLE NO. 3. DIAMETERS WITH CORRESPONDING ANGLES OF THREADS.

TO FIND ANGLE A $\frac{\text{LEAD}}{\text{CIRCUMFERENCE OF SCREW}}$ = TANG. OF ANGLE



A = ANGLE
D = DIAMETER OF SCREW
L = LEAD
P = PITCH OR THREADS PER INCH

THREADS PER INCH	DIAMETERS WITH CORRESPONDING ANGLES																					
	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1	1 1/4	1 1/2	1 3/4	2	2 1/2	3	3 1/2	4	4 1/2	5	5 1/2	6
100	2° 55′	1° 27′	58′	44′	22′	15′	11′	09′	07′	06′	05′									
80	3 39	1 49	1° 13	55	27	18	14	11	09	08	07	05′								
64	4 31	2 17	1 31	1° 08	34	23	17	14	12	10	08	07	06′	05′						
56	5 12	2 36	1 44	1 18	39	26	19	16	13	11	10	08	07	06	05′					
48	6 04	3 02	2 02	1 31	46	30	23	18	15	13	12	09	08	07	06	05′				
40		3 40	2 31	1 49	55	37	27	22	18	16	14	11	09	08	07	06	05′			
36		4 02	2 42	2 02	1° 00	40	30	24	20	17	15	12	10	09	08	07	06	06′	05′	
32		4 33	3 02	2 17	1 08	46	34	27	23	19	17	14	12	10	09	08	07	06	06	
30		4 51	3 14	2 27	1 13	49	37	29	24	21	18	15	12	11	09	08	07	07	06	
28			3 28	2 36	1 18	52	39	31	26	22	19	16	13	11	10	09	08	07	07	
26			3 44	2 48	1 24	56	42	34	28	24	21	17	14	12	11	09	08	08	07	
24			4 03	3 02	1 31	1° 00	46	37	30	26	23	18	15	13	11	10	09	08	08	
22			4 25	3 19	1 39	1 06	50	40	33	28	25	20	17	14	13	11	10	09	08	
20				3 38	1 49	1 13	55	44	37	31	27	22	18	16	14	12	11	10	09	
18				4 03	2 02	1 21	1° 00	49	40	35	30	24	20	17	15	14	12	11	10	
16				4 33	2 17	1 31	1 08	55	46	39	34	27	23	20	17	15	14	12	11	
14				5 12	2 36	1 44	1 18	1° 01	52	44	39	31	26	22	19	17	15	14	13	
13				5 36	2 48	1 52	1 24	1 08	56	48	42	34	28	24	21	18	17	15	14	
12				6 04	3 02	2 01	1 31	1 13	1° 00	52	46	37	30	26	23	20	18	16	15	
11				6 38	3 19	2 12	1 39	1 20	1 06	56	50	40	33	28	25	22	20	18	17	
10				7 16	3 38	2 26	1 49	1 27	1 13	1° 02	55	44	37	31	27	24	22	20	18	
9				8 08	4 04	2 42	2 02	1 38	1 21	1 10	1° 00	49	40	35	30	27	24	22	20	
8				9 08	4 34	3 02	2 17	1 49	1 31	1 18	1 08	55	46	39	34	30	27	25	23	
7					5 12	3 28	2 36	2 01	1 44	1 28	1 18	1° 01	52	44	39	35	31	28	26	
6					6 04	4 02	3 02	2 26	2 01	1 44	1 31	1 13	1° 00	52	46	41	37	33	30	
5½					6 38	4 24	3 19	2 40	2 12	1 52	1 39	1 20	1 06	56	50	45	40	36	33	
5					7 16	4 52	3 38	2 54	2 26	2 04	1 49	1 27	1 13	1° 02	55	49	44	40	37	
4½					8 08	5 24	4 04	3 16	2 42	2 20	2 02	1 38	1 21	1 10	1° 00	54	49	44	40	
4					9 08	6 04	4 34	3 38	3 02	2 36	2 17	1 49	1 31	1 18	1 08	1° 01	55	50	46	
3½						6 56	5 12	4 02	3 28	2 56	2 36	2 01	1 44	1 28	1 18	1 08	1° 01	56	52	
3¼						7 28	5 36	4 32	3 44	3 12	2 48	2 16	1 52	1 36	1 24	1 16	1 08	1° 02	56	
3						8 04	6 04	4 52	4 02	3 28	3 02	2 26	2 01	1 44	1 31	1 22	1 13	1 06	1° 00	
2⅞						8 26	6 20	5 06	4 13	3 37	3 10	2 33	2 07	1 48	1 35	1 25	1 16	1 09	1 03	
2¾						8 48	6 38	5 20	4 24	3 44	3 19	2 40	2 12	1 52	1 39	1 29	1 20	1 13	1 06	
2⅝						9 20	6 56	5 34	4 40	3 58	3 28	2 47	2 20	1 59	1 44	1 33	1 24	1 16	1 10	
2½							7 16	5 48	4 52	4 08	3 38	2 54	2 26	2 04	1 49	1 38	1 27	1 20	1 13	
2⅜							7 42	6 10	5 08	4 24	3 51	3 05	2 34	2 12	1 56	1 44	1 33	1 25	1 17	
2¼							8 08	6 32	5 24	4 40	4 04	3 16	2 42	2 20	2 02	1 50	1 38	1 29	1 21	
2							9 08	7 08	6 04	5 12	4 34	3 38	3 02	2 36	2 17	2 03	1 49	1 40	1 31	

Machinery, N. Y.

was very dirty and Sam mixed up a new lot, and told the boys to dip the work well. The first lot came out of the bath in fairly good shape, a few pieces peeled a trifle, but the next lot came O. K., and Sam had no trouble after that. He stayed until the lock shop was rebuilt and then went back to his old job. Sam told me this himself, and while he has his faults, breaking the ninth commandment is not one of them.

A. P. PRESS.

A CRITICISM.

Editor MACHINERY:

In his article, "An example of worm gearing" in last issue of MACHINERY, Mr. Perrigo finally gets to the point where he shows us his method of sizing worm gears, and the proofs that he advances to show its superiority over other methods are certainly startling, for instead of comparing the behavior of worm gears constructed by his and by other methods, he simply informs us that his worm gears gave satisfaction, perhaps in spite of his method for sizing them. I am not prepared to say that there is nothing in Mr. Perrigo's method, but after reading his proofs I have a strong inclination to think so.

The angle of thread of the worm is likely to have a far greater effect on the behavior of worm gearing than slight differences in the size of the worm gear. If we had been informed about the diameter of the worm, lead or number of threads to the inch, number of revolutions per minute, pressure, etc., we would have been in a position, perhaps, to learn something from the article.

In Brown & Sharpe's "Practical Treatise on Gearing," the method of assuming the pitch diameter of the worm gear from the point where the pitch line of the worm intersects the center line of the worm and worm gear is favored, and adding 2 × addendum for finding the throat diameter. For worm gears with less than 30 teeth, they give a rule for finding throat diameter which, written as a formula, would take the form:

$T = .937D + 4S$, in which

T = throat diameter,

D = pitch diameter,

S = addendum.

Now for comparison let us see what size of worm gear or throat diameter T we get by the first and second methods, and by Mr. Perrigo's method, for a worm gear with 28 teeth, 1.25-inch circular pitch, $S = 0.4$ inch.

1. $T = \dfrac{28 \times 1.25}{\pi} + 2 \times S = 11.14 + 2 \times .4 = 11.94$ inches.

2. $T = .937D + 4S = .937 \times 11.14 + 4 \times .4 = 12.05$ inches.

3. Making a layout according to Mr. Perrigo's method, we will find that $T = 11\frac{5}{8}$ inches, thus making T considerably less than by either of the other methods. In this layout the pitch diameter of worm was assumed to be 3.609 inches. In absences of more information about the relative merits of Mr. Perrigo's method, I hold that the first two methods are more reliable.

E. KWARTZ.

Chicago Heights, Ill.

Reply by Mr. Perrigo.

Editor MACHINERY:

Mr. E. Kwartz goes to considerable trouble to criticise my article on worm gears in the June issue, and to tell us several things about worm gear methods that many mechanics are already familiar with, and which many of them use. While he favors other methods in preference to the one given by me, it will be noticed that he does not present any actual evidence of their practical value. The doctors have a saying that "an ounce of prevention is worth a pound of cure," and I want to remind our Chicago friend that an ounce of success is worth a pound of theory, and also, that his fling that the worm gears on the machine referred to "give satisfaction in spite of the method of sizing them," is in my opinion a rather cheap substitute for argument.

Mr. Kwartz further says: "The proofs he advances to show its superiority over other methods are certainly startling, for instead of comparing the behavior of worm gears constructed by his and by other methods he simply informs us that his worm gears give satisfaction," which is only another way of saying that I have simply told the truth. In comparing the two methods discussed, Mr. Kwartz says that my method makes the throat diameter "considerably less," which is neither very definite, nor very conclusive. He concludes thus: "In the absence of more information about the relative merits of Mr. Perrigo's method, I hold that the first two methods are more reliable." If Mr. Kwartz had written me for information in this matter, I would have been pleased to have given him the information desired on this point, or any other relating to mechanical engineering that I could favor him with so that he might present a reasonable argument against an actual fact.

The gist of the matter is just this. I prefer to use a method that I have found by years of practice to result in success. I have made the statement of an actual and easily established fact. If Mr. Kwartz, or any other mechanic desires to use any other method he has a perfect right to do so. Let him have and enjoy his own opinion, but meantime I shall hold mine as long as I have the argument of practical success to back it.

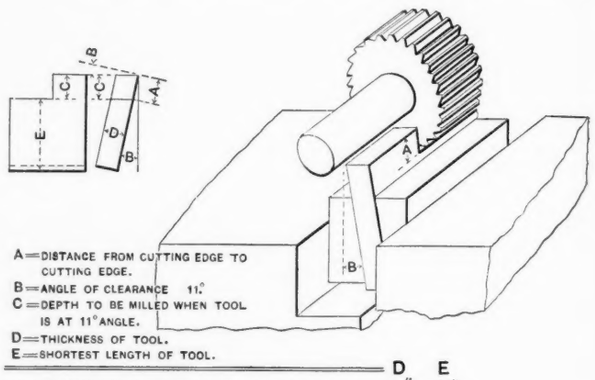
OSCAR E. PERRIGO.

Neponset, Mass.

MILLING FLAT FORMING CUTTERS.

Editor MACHINERY:

The cut and tables of dimensions attached show how we mill flat forming cutters so as to compensate the angular cut necessary for clearance; in other words, how the cutters are measured so that they will cut the actual dimensions required of the screw machine product. The sketches and data are gotten up for the shop in the form of a sheet, 9 x 12 inches, headed "Flat Forming Cutters: Rule for Milling." Following is the table at the bottom of the sheet, giving, for the inch and fractions of the inch, the differences between the distance A and the actual depth of the cut C:


A=DISTANCE FROM CUTTING EDGE TO CUTTING EDGE.
B=ANGLE OF CLEARANCE 11°
C=DEPTH TO BE MILLED WHEN TOOL IS AT 11° ANGLE.
D=THICKNESS OF TOOL.
E=SHORTEST LENGTH OF TOOL.

	D	E
SMALL PRATT & WHITNEY SCREW MACHINE & B. & S.	1/4"	3/4"
1 3/8"—WARNER & SWASEY " "	3/8"	1 5/8"
2 "—WARNER & SWASEY " "	3/8"	1 3/8"
JONES & LAMSON " "	1/2"	2 "

Milling Flat Forming Cutters.

$C = A$ minus .01837 for every inch;

$C = A$ minus .00183 for every 1-10 inch;

$C = A$ minus .00018 for every 1-100 inch;

$C = A$ minus .00229 for every 1-8 inch;

$C = A$ minus .00114 for every 1-16 inch;

$C = A$ minus .00028 for every 1-64 inch;

Detroit, Mich. W. E. GOULD.

* * *

LOCOMOTIVES AS FOG-MAKERS.

An engineer asserts that the London fogs are caused largely by the discharge of steam into the air from the 300 or more locomotives operating in the London area. Averaging each engine at 500 horsepower, or a total of, say, 150,000 horsepower, and adding 500,000 horsepower from other steam-generating sources, gives a total of 650,000 horsepower of steam discharged into the atmosphere. One steam plant of 2,000 horsepower, the engineer figures, will discharge into the air twenty tons of steam per hour, or sufficient to produce a fog twenty feet thick and one mile square; and what plants with a capacity of 650,000 horsepower can do in the way of fog production may, therefore, be easily estimated. The proposed remedy is to convert this waste steam into electric power.—*Railway Age.*

HAND TAPS WITH STANDARD THREADS.

H. D.

In the following is a complete set of empirical formulas and a table for well proportioned hand taps, insuring uniformity as well as a neat appearance.

It will be noted that the formulas are based upon the diameter of tap, this being the most convenient working factor, as, of course, the diameter is always given from the beginning. At the first glance an observer might infer that the working factor ought to be the number of threads per inch, but as that number in all standard systems is dependent upon and stands in a certain relation to the diameter this latter factor is just as correct to work from, and gives simpler and more universal formulas.

It is obvious that formulas cannot be made up that would suit the whole range of diameters from the very smallest up to the very largest, and, therefore, it has been necessary to divide the series in two groups in order to obtain correct proportions, the one group including taps from 3/16 inch up to 1 inch diameter; the second from 1⅛ up to 4 inches diameter.

In the following formulas:

a = the total length of tap,
b = the length of thread,
c = the length of shank,
d = the diameter of tap,
e = the diameter of shank,
f = the size of the square,
g = the length of the square.

TABLE OF HAND TAPS WITH STANDARD THREADS.

Diam. of Tap.	Total Length.	Length of Thread.	Diameter of Shank.			Size of Square.	Length of Square.
			U. S. Standard Thread.	Standard V-thread.	Whitworth Standard Thread.		
3/16	2 1/4	7/8	3/16	3/16	3/16	1/8	3/16
1/4	2 1/2	1	1/4	1/4	1/4	3/16	1/4
5/16	2 15/16	1 1/4	.28	.25	.28	3/32	5/16
3/8	3 3/8	1 1/2	.39	.34	.38	9/32	7/16
5/16	3 13/16	1 3/4	.49	.45	.49	3/8	9/16
1/2	4 1/4	2	.61	.56	.61	15/32	5/8
7/16	4 1/16	2 1/4	.72	.67	.72	9/16	3/4
1	5 1/8	2 1/2	.82	.77	.82	5/8	13/16
1 1/8	5 1/2	2 5/8	.91	.85	.92	11/16	7/8
1 1/4	5 13/16	2 3/4	1.04	.98	1.04	3/4	15/16
1 3/8	6 1/8	2 7/8	1.13	1.07	1.14	7/8	15/16
1 1/2	6 3/8	3	1.26	1.19	1.26	15/16	1
1 5/8	6 11/16	3 1/4	1.37	1.26	1.34	1	1 1/16
1 3/4	6 15/16	3 1/4	1.47	1.38	1.47	1 1/8	1 1/16
1 7/8	7 1/4	3 5/8	1.59	1.47	1.57	1 3/16	1 1/8
2	7 1/2	3 1/2	1.69	1.59	1.69	1 1/4	1 3/16
2 1/8	7 13/16	3 5/8	1.81	1.72	1.82	1 3/8	1 3/16
2 1/4	8 1/16	3 3/4	1.94	1.84	1.91	1 7/16	1 1/4
2 3/8	8 3/8	3 7/8	2.03	1.97	2.03	1 1/2	1 5/16
2 1/2	8 5/8	4	2.15	2.04	2.15	1 5/8	1 5/16
2 5/8	8 15/16	4 1/8	2.28	2.17	2.28	1 11/16	1 3/8
2 3/4	9 3/16	4 1/4	2.40	2.29	2.36	1 13/16	1 7/16
2 7/8	9 1/2	4 3/8	2.48	2.42	2.48	1 7/8	1 7/16
3	9 3/4	4 1/2	2.60	2.48	2.61	1 15/16	1 1/2
3 1/8	10 1/16	4 5/8	2.73	2.61	2.73	2 1/16	1 9/16
3 1/4	10 5/16	4 3/4	2.85	2.73	2.86	2 1/8	1 9/16
3 3/8	10 5/8	4 7/8	2.95	2.81	2.96	2 1/4	1 5/8
3 1/2	10 7/8	5	3.08	2.94	3.08	2 5/16	1 11/16
3 5/8	11 3/16	5 1/8	3.20	3.06	3.20	2 3/8	1 11/16
3 3/4	11 7/16	5 1/4	3.29	3.15	3.33	2 7/16	1 3/4
3 7/8	11 3/4	5 3/8	3.42	3.27	3.44	2 9/16	1 13/16
4	12	5 1/2	3.54	3.40	3.55	2 5/8	1 13/16

For taps from 3/16 to 1 inch diameter the following formulas are to be used:

$a = 3.5d + 1\frac{5}{8}$ inches,
$b = 2d + \frac{1}{2}$ inch,
$c = 1.5d + 1\frac{1}{8}$ inch,
e = root diameter of thread —.01 inch.
$f = \frac{3}{4}e$.
$g = \frac{3}{4}$d + 1/16 inch.

For taps from 1⅛ to 4 inches diameter the following formulas are to be used:

$a = 2.25d + 3$ inches,
$b = d + 1\frac{1}{2}$ inches,
$c = 1.25d + 1\frac{1}{2}$ inches,
e = root diameter of thread —.02 inch,
$f = \frac{3}{4}e$.
$g = d/3 + \frac{1}{2}$ inch.

In the tables the values have been given in even 16ths in cases where the formulas would give values in thousandths inch. This has been done everywhere except in the case of the diameter of the shank, e, where the value, figured from the formula has been given in the nearest smaller one-hundredth inch. The shanks of 3/16 and ¼ inch taps are usually made same as the diameter of the tap.

It is well known that hand taps are made in sets of three, taper, plug and bottoming. They are generally made as follows: The point of the taper tap is turned down to the diameter at the bottom of the thread or a few thousandths smaller, according to the size and pitch of the tap. This acts as a guide and aids in securing a straight tapped hole. This guide ought to be about the length of three or four threads. From the upper end of this guide the thread is tapered until it reaches the full diameter of the tap. The length of this tapered portion is from 6 to 7 threads. The plug and bottoming taps are practically the same, the only difference being that the former has a chamfer of about three threads and the latter of about one thread at the point; the diameter of the straight portion of all the taps in the set is the same.

However, from a critical point of view this way of making taps intended to be used in sets cannot be considered correct, because, as the taps have all the same diameter, the tapered portion of the first tap will have the bulk of the work to do, while the two following taps practically have no work to do except in a case where a full thread is required at the bottom of a hole; but even then the duties of the different taps in the set is rather unevenly distributed.

For this reason it is very obvious that taps intended for use in sets should vary in diameter, so that each tap will have a reasonable amount of work to do; of course, the last tap, being a finishing tap, should have less work to do than the two first ones. This way of making hand taps in sets, although being both for practical and theoretical reasons the only correct and the best way, does not seem to have met with the favor of the tap manufacturers, there being only one leading firm (the Pratt & Whitney Co.) who manufacture hand taps made in this manner.

The principal objection to making hand taps in sets as described above, and the probable cause for their slow introduction, must be, that when using taps of such description the whole set always has to be used, whereas for a short hole to be tapped clear through a piece the taper tap alone will be found sufficient, if the straight portion of the tap is up to the full diameter. And in fact, all three taps, when made all with the same diameter, are seldom used except when a full thread is wanted at the bottom of a hole. However, to cut the full thread of a hole tapped clear through a piece, by the taper tap in one operation places an undue stress on this tap, and will not give as smooth a thread as if the hole had been run through by a set of taps, each of which cuts a fair amount of the thread, by means of the difference in diameter of the different taps.

In this connection it may be well to mention another quite as important factor in the making of hand taps—the fluting. Both a strong, an easily made, and easy working tap is always looked for. These three factors are greatly dependent upon the way the tap is fluted. It has become more and more the practice in manufacturing taps to make a deep straight-sided flute with a small round of about one-eighth the diameter of the tap at the bottom. This, however, although it has its advantages in providing for more chip-room, has some very grave disadvantages. The tap will crack more easily in hardening, will not cut as freely, and is not as strong as one fluted with a regular convex cutter.

In regard to the number of flutes there is some difference in opinion. There are those who consider four flutes the proper number to use on all sizes of hand taps with the land about one-fourth the diameter of the tap. However, on large taps the land will be rather wide if made according to this rule, and better results will be obtained by increasing the number of flutes for those diameters which are larger than 2 inches.

* * *

Sir Wilhelm Siemens announces a new use of vanadium in an electric glow lamp.

SHOP KINKS.

A DEPARTMENT OF PRACTICAL IDEAS FOR THE SHOP.
Contributions of kinks, devices and methods of doing work are solicited for this column. Write on one side of the paper only and send sketches when necessary.

A POINTER ON FILING.

In order to make it easy to see the part being filed on work, such as backing off the teeth of small milling cutters and of a like nature, I have been in the habit of using copperas or blue vitriol solution, which as we all know, gives a rather light copper color, but have recently adopted the idea of drawing the piece to be filed to a dark blue over a Bunsen burner. It is needless to say the contrast between the dark blue and the filed parts is very marked—it can't be beat.

Chicago, Ill. R. A. LACHMANN.

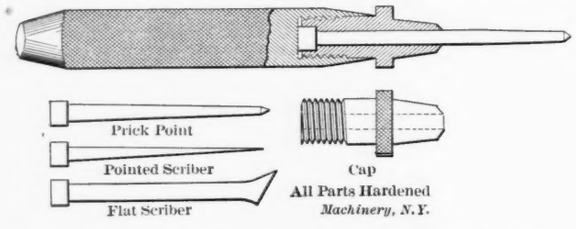
APPLYING ASBESTOS.

Asbestos is of such common use around the steam plant that a knowledge of quick and neat methods of putting it on is sure to be of service. The following meets this requirement: Mix the asbestos thoroughly with water so as to make it plastic; put first coat on rough with pointed trowel about ½-inch thick; after this is dry put on a second coat ¾-inch thick, and straighten down with large trowel. The second coat should be wired by winding with No. 16 annealed wire, the winding to be about 3 inches apart; or in case of flat surfaces, by binding it with horizontal wiring. Apply third coat thick enough to cover wire and make smooth surface. A bag weighing 100 pounds will cover about 40 square feet. JAMES A. PRATT.

Howard, R. I.

PRICK-PUNCH WITH INTERCHANGEABLE SCRIBER POINTS.

In the tool shown herewith, we have a departure from the simple construction of the ordinary prick-punch inasmuch as it is made so that a number of interchangeable points can be substituted at will. While such a tool is one that will be appreciated by the traveling mechanic as it helps him in keeping the weight of his kit within portable limits, its principal advantage lies in that the point can be kept sharp easier since



the cross-section does not increase much up to the holder. When the point is ground away, another can be substituted without bothering the tool-dresser. Moreover, it is something different from the ordinary tool, and that is a distinction that many men like. C. C. ARRAS.

Buffalo, N. Y.

HACK SAW KINKS.

We all know how difficult it is at times to saw straight in the regular run of power hack saws, but to overcome the difficulty is a very simple matter if we only knew how, and here's the how. For instance, we have a piece of mild steel, say ¾-inch thick by 3 inches wide to saw. We know it "goes" much quicker to saw it edgewise than flat, so we want to keep on doing so, but first of all start the cut on the flat side and let the saw make a few strokes, enough to go into the metal about ⅛ inch, then set up on edge so the saw is directly over the cut first made and, following along the lines that "water won't run up hill," but will follow the path of least resistance, same as lightning, steam, electricity, etc., the saw will be guided by the shallow saw cut. Another point to be observed in the use of hack saws for either power or hand frames is preventing further damage to the saw when a few teeth are broken out. Stop right there and grind the teeth on each side of the break on a gradual slant in toward the opening. This will make the saw almost as good as before and prevent ripping out more teeth. R. A. LACHMANN.

Chicago, Ill.

A PIPE FISHING JOB.

I was once called upon to devise a scheme by which a piece of 1½-inch pipe about 15 feet long, which had accidentally fallen to the bottom of an artesian well, could be abstracted. The well mentioned was of 3-inch pipe and 85 feet in depth, the last ten feet being driven in quicksand, which had sifted in through the strainer in considerable amount. This, in conjunction with the additional piece of pipe which had forced its way into it several feet, was checking the supply of water, and as the boiler depended entirely upon the water supply from this well (there being no running water in the village), matters became serious. After considerable meditation the following method was decided upon:

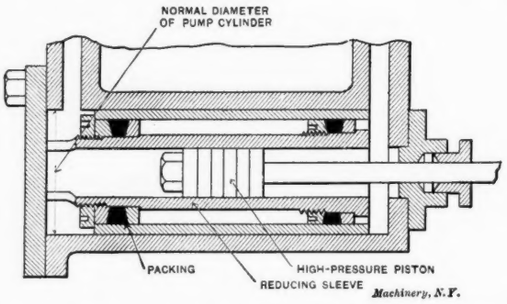
Four lengths of ½-inch gas pipe, a 1½-inch to ½-inch reducing coupling, and a piece of thoroughly seasoned pine were procured, the latter being turned into a long, slightly tapering plug to fit the 1½-inch pipe. One end was securely fastened to the reducer coupling, and the other end was sharpened to a point. As soon as one length of pipe had been lowered into the well another was screwed on as quickly as possible, and when at last the desired goal was reached the plug was crowded firmly into the pipe and given time to swell. It was necessary to use considerable force to start the pipe out of the sand but it stood the test without the slightest slip.

New Haven, Conn. OTIS D. STORER.

* * *

REDUCING SLEEVE FOR PUMP CYLINDERS.

A simple method by which an ordinary boiler feed pump may be converted into a high-pressure pump for testing purposes is shown in the illustration. It consists of a reducing sleeve which is inserted into the pump cylinder and which is provided with suitable stuffing-boxes to prevent the water by-passing from one end of the cylinder to the other. The regular



pump piston is removed, of course, and a smaller one substituted of the proper size to fit the reducing sleeve. The scheme is one that has met with some favor in marine service, as it enables an engineer to test any of his boilers with one of the regular feed pumps and it saves the labor necessary to work hand test pumps.

* * *

The effect of the traction wheels of electric motor cars seems to be quite different than from that of the drivers of locomotives. This was alluded to by Mr. H. M. Steward in a paper read before the New England Street Railway Club, October 27, 1904. An engine driver can be spinned or turned in the same place for some time without injuring the rail, the only result being that the rail becomes slightly burned and hardened at the point where the driver was turning. But in the experience of the Boston Elevated Railway, if the motor wheels of the car are kept turning for any length of time they will actually cut a 85-pound rail in two. There have been cases where trains have been stalled on grades, and in endeavoring to start the train, the motor wheels have been revolved in the same place for a short time with a result that deep holes had been cut into the rails. When this has occurred the trains could not have started by their own power, but were obliged to remain where they were until pulled or pushed out of the holes.

SHOP RECEIPTS AND FORMULAS.

Practical Tried Receipts—those known to be Good—are Solicited.

SOMETHING ABOUT THIS DEPARTMENT.

The object of this department is to publish practical tried shop receipts that are known by the contributor to be O. K. It is one thing to make a collection of receipts and formulas, and quite another thing to be sure that each one is good. With the coöperation of our readers we hope to put on record during the next year or two the best collection that has ever been published, inasmuch as each is expected to have the personal recommendation of its contributor. Remember that we pay for all accepted contributions, and that we pay more for receipts than for any other class. Requests for specific receipts that are not generally known will appear in the "How and Why" department.—EDITOR.

CEMENT FOR IRON.

Six parts of white lead, 6 of sulphur and 1 of borax, thoroughly mixed and wetted with strong sulphuric acid makes a very strong cement. DAVID MELVILLE.

Detroit, Mich.

MIXTURE FOR FIRE CEMENT.

To make a fire cement use 100 parts fire clay, wet; 3 parts black oxide manganese; 3 parts white sand; and ½ part powdered asbestos. Thoroughly mix, adding sufficient water to make a smooth mortar. C. E. MINK.

Syracuse, N. Y.

TO COPPER BRASS FOR LAYING OUT WORK.

To apply a copper coloring upon brass for laying out work, put a few drops of the ordinary coppering solution upon the brass and then dip a piece of iron or steel into the solution and touch the brass. OSCAR J. BEALE.

Providence, R. I.

TO PREVENT BABBITT METAL OR LEAD FROM EXPLODING.

Before pouring the babbitt metal, throw in a piece of resin the size of a walnut and allow it to melt. If the bearings to be lined with babbitt are warmed before pouring, the metal will run better, thus insuring a better job.

Schenectady, N. Y. R. B. CASEY.

VARNISH FOR STEEL.

A good varnish for steel may be made by dissolving 10 parts of clear grains of mastic, 5 parts of camphor, 15 parts of sandarach, and 5 parts of elemi in a sufficient quantity of grain alcohol. Apply the varnish without heat. JOS M. STABEL.

Rochester, N. Y.

TO SOFTEN HARD CAST IRON FOR DRILLING.

Heat to a cherry red, having it to lie level in the fire. Then with a pair of cold tongs put on a piece of sulphur a little less than the size hole to be drilled. This will soften the iron entirely through, providing it is not too thick.

Dayton, O. O. E. VORIS.

TO REMOVE BLUING FROM TEMPERED STEEL.

Plunge the blue hot article into a bath of sulphuric acid 1 part, water 16 parts; then into a bath of lime and water (to neutralize the acid) and rub it off quickly with a dry cloth and Vienna lime. The result will be a most beautiful polish.

Angelica, N. Y. F. H. JACKSON.

POLISH FOR BRASS—POLISH FOR STEEL.

A good polish for brass is made by putting 2 ounces of sulphate of nickel and 2 ounces of nitric acid in an open vessel and allowing them to mix thoroughly. Then add water.

To make a polish for steel dissolve 2 ounces each of oxalic acid, pumice stone, ammonia, and whiting in a quart of water.

New York. HERMAN JONSON.

CEMENT FOR HOLES AND CRACKS IN CASTING.

A cement for holes and cracks in castings is made of 6 parts red lead, ground in oil; 3 parts white lead, ground in oil; 2 parts black oxide of manganese; 1 part silicate of soda; ½ part litharge. Mix and use as a putty. To preserve from hardening put in a vessel and cover with water.

Syracuse, N. Y. C. E. MINK.

ETCHING FLUID.

Mix 1 ounce of sulphate of copper, ½ ounce of alum and ½ teaspoonful of salt reduced to a powder, with 1 gallon of vinegar and 20 drops of nitric acid. This fluid may be used for eating deeply into the steel or metal, or for imparting a beautiful frosted appearance to the surface according to the time it is allowed to act. DAVID MELVILLE.

Detroit, Mich.

LUBRICANT FOR DRAWING DIES

The following mixture has given very good results as a lubricant on drawing dies when drawing sheet metal: Boil together until thoroughly mixed, 1 pound of white lead, 1 quart of fish oil, 1 pint of water, and 3 ounces of black lead. Apply to the sheet metal with a brush before it enters the dies.

Rochester, N. Y. JOS. M. STABEL.

THE USE OF BRASS WIRE IN BRAZING.

In place of spelter use wire or rod brass and boracic acid as a flux. Anneal the end of wire or rod by heating while the joint is getting hot, and after dipping the rod into boracic acid, apply to the joint; the rod melting at the end will flow into the joint. After the joint is cooled, submerge in hot soda water; this will take off every particle of acid, leaving only the brass to be filed off. F. H. JACKSON.

Angelica, N. Y.

FOR BLUING SMALL BRASS AND STEEL ARTICLES BY IMMMERSION.

To blue small brass articles by immersion, use chloride of antimony, 1 ounce; water, 20 ounces; hydrochloric acid, 3 ounces. Place the solution in an earthen jar and suspend the piece in this bath until blue, then wash and dry in sawdust. The pieces should be warmed first.

To blue steel without heat, apply nitric acid; wipe off the acid clean, oil and burnish. L. E. MUNCY.

Syracuse, N. Y.

TRANSMISSION ROPE DRESSING.

A good transmission rope dressing is made by melting together 450 pounds of tallow, 33 pounds resin, 150 pounds beeswax, 20 pounds pine tar, 14 pounds lampblack, and 15 pounds tobacco tin-foil. Pour the mixture in molds to make stock 2½ inches in diameter, and 11 inches long, weighing about 3 pounds each. Use one for about 400 feet of one-inch rope.

New York. HERMAN JONSON.

VARNISH FOR CAST IRON PATTERNS.

For small cast iron patterns the following is a very satisfactory method of varnishing. Apply boiled linseed oil to the iron, the pattern being heated to a temperature that will just char or blacken the oil; the oil appears to enter the pores of the iron, and after such an application the metal resists rust and corrosive agents very satisfactorily. JAMES A. PRATT.

Howard, R. I.

CASE-HARDENING CAST-IRON.

To successfully caseharden cast-iron, the pieces to be hardened should be heated to a red heat, then rolled in a composition of equal parts of prussiate of potash, sal-ammoniac and saltpeter. All pulverized and thoroughly mixed. Every part of the casting must be covered by the composition before plunging (red-hot) into a bath of 2 ounces prussiate of potash and 4 ounces sal-ammoniac to each gallon of cold water. A.

TO BRONZE YELLOW BRASS.

To produce a bronze finish on rough yellow brass castings, mix equal parts of nitric acid, sulphuric acid and water; the nitric acid and water should be mixed first and the sulphuric acid added slowly. Dip the yellow brass castings into boiling water a moment, then in the acid solution, then quickly back into the boiling water, and rinse thoroughly in clean water. Dry in pine sawdust. The castings must be perfectly free from soldering solutions, etc., or stains are liable to appear. This method gives a finish similar to gas fixtures, etc., and may be rendered very permanent by coating with a transparent lacquer. I. W. ANTANO.

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

The following questions are referred to our readers:

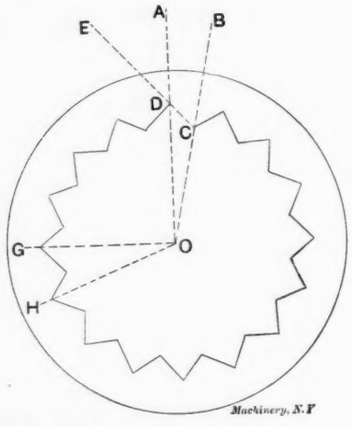
50. A. R. T.—How are hacksaw blades hardened so that the teeth are hard and the backs left soft?

51. I. T. O.—I want a good process for tinning wire goods like the wire baskets for office desk use; the use of pure tin has been unsatisfactory with me because of tne expense of pure tin, which costs about 33 cents a pound. Perhaps if I could get a thin even coating all over the goods it would be cheap enough, but this I have been unable to secure so far. Can a mixture of tin and lead be used for this purpose successfully?

---

52. L. E. M.—Please give me a rule for figuring the angle to make a V tool to plane notches in the inside of a hole or ring, and leave points of a given angle. I would like a rule that will cover this, irrespective of number of divisions or diameter of circle.

A.—The sketch represents a ring, on the inside of which teeth have been cut as shown. Since the tool is to be shaped to cut out the notches between the teeth, the angle of the tool must be equal to the angle between the sides of any one of the notches. Draw the radial line *OA*, bisecting one of the notches, and the radial line *OB* bisecting one of the teeth.



Then $ECB = \frac{1}{2}$ angle of tooth, $ODC = \frac{1}{2}$ angle of tool, and $AOB = \frac{1}{2}$ center angle. By center angle is meant the angle between radial lines drawn from the center of the ring to corresponding points of two adjacent teeth, as for example, angle *GOH*. By a principle of geometry, the exterior angle *ECB* of the triangle *ODC* equals the two opposite interior angles of the triangle.

Hence, angle *ECB* = angle *ODC* + *DOC*, or

angle *ODC* = angle *ECB* — angle *DOC*.

Multiplying by 2,

2 × angle *ODC* = 2 × (angle *ECB* — angle *DOC*), or

$$\text{Angle of tool} = \text{angle of tooth} - \text{center angle} = \text{angle of tooth} - \frac{360}{\text{number of teeth}}.$$

Example: Required, angle of tool when angle of tooth = 90 degrees and the circle is to be cut with 60 teeth.

$$\text{Angle of tool} = 90 - \frac{360}{60} = 90 - 6 = 84 \text{ degrees.}$$

---

53. E. J. B.—Is there any way in which a wattmeter can be made to creep or not creep, at the will of an operator at a central station, and when all the switches at the receiving station are cut out, and no motors running. I have one in my charge that has been creeping every night and Sundays for six weeks, yet when under investigation by myself and officials of the power company it does not creep at all, and under exactly the same conditions.

Answered by Mr. Wm. Baxter, Jr.

Any one who is thoroughly familiar with the wiring from the generators or switchboard to the receiving station, can make connections through which the wattmeter can be made to creep. The two sketches, Figs. 1 and 2, show how it can be done, the first arrangement being that required if the consumer is connected with the line through a single pole switch, and the second if the connection is through a double pole switch. The vertical line *a a* separates the station end from the consumer end, the latter being on the right side of the line. In Fig. 1 if the consumer's circuit is grounded as at g' and the station is grounded at g, with a switch to make and break the ground connection, then when this switch is closed, the circuit will be completed, and if there is a path in the consumer's circuit as indicated by *R*, current will pass in an amount that will depend upon the resistance at *R*. It is not customary to use a single pole switch to connect a consumer with the lines, therefore, the case shown in Fig. 2 is more likely to be met. In this case the arrangement is the same as in the first sketch with the exception that a connection is made around the upper switch *s* by means of the wire *b*. Any such condition as is above indicated, can be detected by disconnecting your circuit from the line completely and then testing it for grounds and for connection with the station circuits.

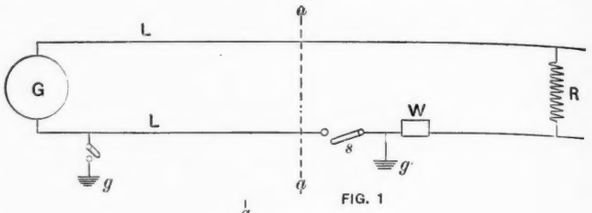


FIG. 1

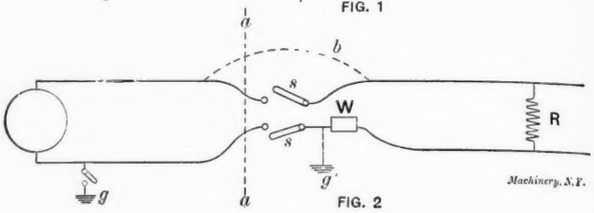


FIG. 2

---

54. A. H. N.—What is the best practice for relieving the teeth of hand taps? Some toolmakers, I find, are firm believers in relieving the teeth; others claim that it is not only unnecessary, but harmful.

A.—Practice differs quite widely; some manufacturers, we believe, still adhere to the older system of relieving the teeth by backing off, *i. e.*, making the tooth narrower and the thread channels deeper at the heel than at the cutting edge; others have abandoned this method and now make hand taps which work very successfully without having any relief whatever, in the ordinary meaning of the term. The newer practice is to cut the thread slightly deeper at the shank end of the thread than at the point, or, in the case of a taper tap, where the thread reaches full diameter. The taper at the root of the thread is usually about 0.001 inch per inch of length, but, in no case, should it be more than 0.003 or 0.004 inch in the length of tap. Thus, a one-inch plug end tap having three inches of thread would be about 0.003 inch less diameter, measured at the shank than at the point. With a taper tap having one-inch taper and two inches of full thread, the shank end would measure about 0.002 inch less at the root of the thread than at the full diameter. This practice is very simple and is claimed to give uniformly excellent results. It works particularly well on taps that have to be backed out, as it removes the liability of chips wedging in on the back of the teeth and breaking them, which is a fault with the back-off teeth. The foregoing does not apply to pipe, boiler, machine and staybolt taps, as these should be relieved in the angle of the thread as well as at the top.

---

55. A. L. B.—I recently made a sketch for laying out a right angle channel on the end of a steel block, 9 inches square, so that when planed out the cross-section shape would be like that shown in the cut herewith. One side of the block was to be 8 inches and the other 9 inches high after planing, and the thickness of metal at the bottom of the channel was to be 5 inches. The two flats *Aa* and *Cc*, were to be of the same width, and the sides of the channel, *AB* and *BC*, of the same length. Although this was all the data given it was a simple matter to make the sketch to fill the bill, but when I tried to calculate

the distances AB and y, to verify the layout dimensions, I got stuck. Is there a simple way of finding these distances by calculation, with the given data?

A. One solution, which can scarcely be called simple, is as follows: Add to the diagram, Fig. 1, the lines AD, DB, BE and EC, drawn parallel to the sides of the block; also the lines AF and AC as shown in Fig. 2. Then by subtracting the 5-inch dimension, representing the distance from the bottom of the channel to the base, from the respective sides, it is found that EC is 4 inches, and AD 3 inches. We also discover

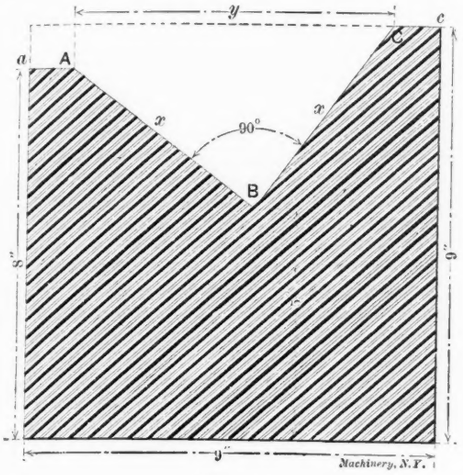


Fig. 1.

that AF is 1 inch, since the short side after planing measures 8 inches and the original height was 9 inches. Triangles AFC, ADB, BEC and ABC are right angled, hence the sums of the squares of the sides of each equal the squares of their respective hypotenuses. Therefore,

$$\overline{AB}^2 = \overline{BD}^2 + 3^2, \text{ and}$$
$$\overline{BC}^2 = \overline{BE}^2 + 4^2$$
$$\overline{AB}^2 - 9 = \overline{BD}^2$$
$$\overline{BC}^2 - 16 = \overline{BE}^2$$
$$\sqrt{\overline{AB}^2 - 9} = BD$$
$$\sqrt{\overline{BC}^2 - 16} = BE$$

Since $AB = BC$, and $DB + BE = FC$, it follows that

$$\sqrt{\overline{AB}^2 - 9} + \sqrt{\overline{AB}^2 - 16} = FC$$
$$\overline{AB}^2 - 9 + 2\left(\sqrt{\overline{AB}^2 - 9}\ \sqrt{\overline{AB}^2 - 16}\right) + \overline{AB}^2 - 16 = \overline{FC}^2$$

In the triangle ABC

$$\overline{AB}^2 + \overline{BC}^2 = \overline{AC}^2, \text{ but}$$
$$\overline{AC}^2 = \overline{FC}^2 + \overline{AF}^2, \text{ hence}$$
$$2\,\overline{AB}^2 = \overline{FC}^2 + 1$$
$$2\,\overline{AB}^2 - 1 = \overline{FC}^2, \text{ therefore,}$$

$$2\,\overline{AB}^2 - 1 = \overline{AB}^2 - 9 + 2\left(\sqrt{\overline{AB}^2 - 9}\ \sqrt{\overline{AB}^2 - 16}\right) + \overline{AB}^2 - 16$$
$$2\,\overline{AB}^2 - 1 = 2\,\overline{AB}^2 - 25 + 2\left(\sqrt{\overline{AB}^2 - 9}\ \sqrt{\overline{AB}^2 - 16}\right)$$
$$2\left(\sqrt{\overline{AB}^2 - 9}\ \sqrt{\overline{AB}^2 - 16}\right) = 24$$
$$\overline{AB}^4 - 25\,\overline{AB}^2 = 0$$
$$\overline{AB}^4 - 25\,\overline{AB}^2 + 156\tfrac{1}{4} = 156\tfrac{1}{4}$$
$$\overline{AB}^2 - 12\tfrac{1}{2} = 12\tfrac{1}{2}$$
$$\overline{AB}^2 = 25$$
$$AB = 5, \text{ from which it follows that}$$
$$DB = 4, \text{ and}$$
$$BE = 3, \text{ hence}$$
$$FC = 7$$

The width of the flats Aa and Cc is 1 inch

---

56. L. H. W.—In using the rule given in Mr. Woodbury's article in the December, 1904, number of MACHINERY upon the location of the decimal point in slide rule calculations, I find it true with the lower scales. How can it be used with the upper scales?

Answered by Mr. S. E. Woodbury.

In the body of the article a sentence reads: "This method is best used with the C and D scales, but can be used as well with the A and B scales after it is once understood." The application is a little indirect, but possibly an example will make it clear. Multiply 3.5 by 3.1416 on the C and D scales; the result is 11. The slide projects to the left and covers the sign P—, therefore there are $1 + 1 = 2$ digits in the result. Using the A and B scales, set the left index of B to 3.5 in the left of A, and the result 11 is read in the right of A over π on B, that is, the result is read in the other half of the scale from that in which one of the factors was taken. This corresponds to the slide projecting to the left when using the C and D scales, hence there is no correction, and there are $1 + 1 = 2$ digits as before. Without disturbing the slide as set for the last reading, it will be noted that under 3.5 in the right half of the A scale is found the middle index of the B scale, and the result is over π as before. The relation of these scales is identical to that of the C and D scales for the same result. Reversing the example and attempting to divide 11 by 3.1416 we find a similar relation of the scales for division. From these relations we can form the rule for the correction. When the result of multiplication or division is found in the same half of the A scale as that from which a factor or a dividend is taken, there is a correction of -1 or $+1$ respectively. When the result is found in the other half of the A scale from that in which a factor or a dividend is taken, there is no correction. This assumes that only one (though it may be either) half of the B scale is used in any calculation.

The following method of locating the decimal point in calculations performed on the A and B scales may be of interest. Near the 3.5 mark in the left half of the A scale write the

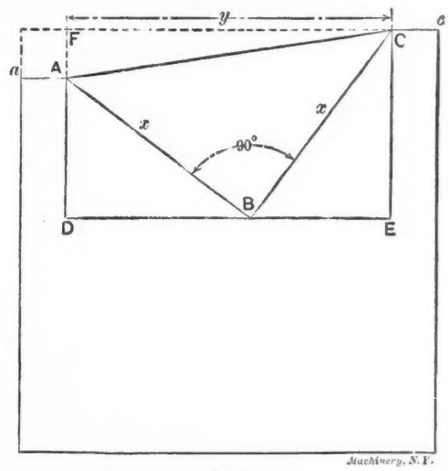


Fig. 2.

sign P—, and near the 3.5 mark in the right half of the A scale write the sign $Q+$. Always take one factor for multiplication in that portion of the scale where the P— is; and, always take the dividend in that portion of the scale where $Q+$ is. Use only one-half of the B scale, reading at the middle index when necessary. If a result of multiplication occurs in the portion of the scale under P—, 1 is to be subtracted from the sum of the digits; if the result occurs in the portion of the scale where the $Q+$ is, there is no correction since $Q+$ does not refer to multiplication. If a result of division occurs in a portion of the scale where $Q+$ is, 1 is to be added; if the result occurs in the portion of the scale where P— is, there is no correction, since P— does not refer to division. I prefer this to any other direct method as applied to the A and B scales, but consider the slightly modified form of the rule for the C and D scales more satisfactory.

* * *

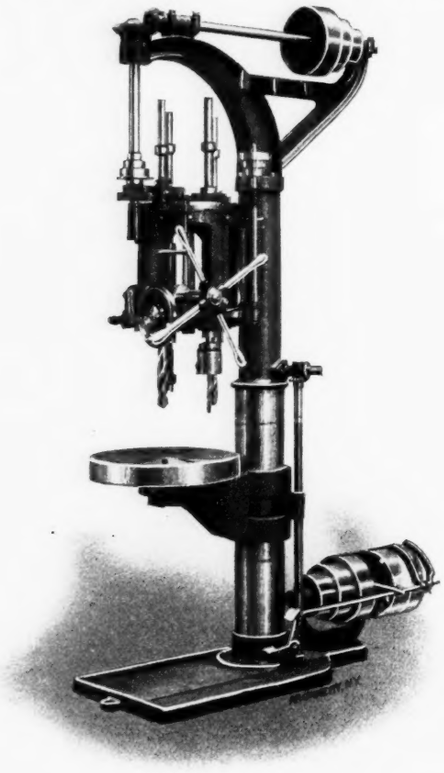
It is suggested that one reason for the extreme secrecy under which certain electric furnace processes are carried out is their simplicity. The extension of ordinary courtesy to visiting engineers in many electro-chemical plants would reveal the processes to such an extent that the visitor could carry away the entire scheme in his head and become a competitor in short time.

MACHINERY AND TOOLS.

A MONTHLY RECORD OF NEW APPLIANCES FOR THE SHOP.

CYLINDER TURRET DRILL PRESS.

A cylinder turret drill press brought out by the National Separator & Machine Co., Concord, N. H., is illustrated herewith. This machine is for drilling, tapping, reaming, etc., on duplicate parts without the necessity of changing the tool or the work. It can be furnished with four spindles as shown in the illustration or with six, and with either power, hand or lever feeds or all of these. The turret carrying the spindles revolves in the outer casing, and is arranged so that when the desired spindle is turned to the front the driving gear connects with the gear on the spindle and causes it to revolve, while the other spindles are left back out of the way and inactive. When the other spindles are to be used the turret is revolved to each in turn. As shown in the cut the same size gear is placed on all the spindles but one, which is geared down for tapping and counterboring. The spindles can be fed to the work directly by the star wheel, by the hand wheel in front, or by power, and can be set to stops to knock off automatically. There is an independent stop for each spindle. All the spindles work to the same point on the platen. A locking device is provided directly in front of the spindle, the turret locking automatically when in position. There is little strain on the turret, as the moment power is applied the strain passes to the frame, the turret acting simply as a guide. The number 1 size of this drill will carry up to 1-inch drills, will swing over the platen 24 inches, and carries four spindles, one being a slow-speed spindle. It is provided with power and lever feed and stops for each spindle.



Cylinder Turret Drill Press.

THE DRAPER GEARED HEAD LATHE.

The Draper 18-inch patent geared lathe is shown in the accompanying half-tone, Fig. 1. An interior view of the head with the cover removed is shown in Fig. 2. The lathe is driven by a wide-faced pulley with a six-inch belt, giving 10 geared speeds to the spindle, obtained by sliding gears operated by levers, as shown in the half-tones. The lathe is made to withstand severe duty, having power either for heavy reduction or fast cutting speed for finishing cuts. The driving shaft is at the back of the lathe, or the top of the interior view, Fig. 2. On this are mounted two sliding gears *G* and *A* operated by the middle lever at the front of the head. These gears mesh in either one of the two gears *H* and *B*, shown on the sleeve running loose on the main spindle. The front countershaft with its four gears, *D*, *K*, *L* and *E*, is operated as is the usual back gear shaft. By throwing out these gears with the eccentric through the handle to the left, Fig. 2, and locking the spindle train with the knob shown protruding from the face of gear *M*, Fig. 2, two rates of speed may be obtained through *A* and *B*, or *G* and *H*. This knob is made accessible to the operator by means of a small plate on top of the cover. With the spindle gears disconnected and the back gear shaft thrown in, there are eight more speeds, making ten in all with the single speed countershaft or the constant speed motor. These eight back geared speeds are obtained by the meshing of either *A* with *B* or *G* with *H*, *C* with *D* or *I* with *K*, and *E* with *F* or *L* with *M*.

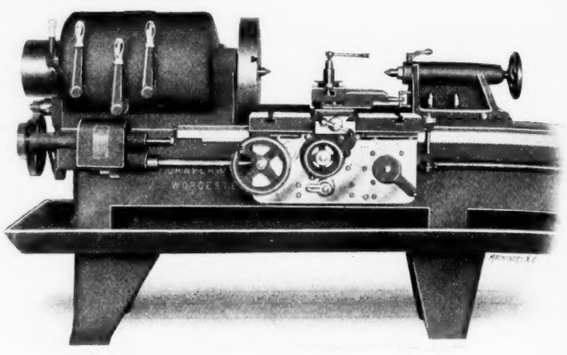


Fig. 1 Draper Geared Head Lathe.

The belt-driven lathe is usually furnished with a two-speed countershaft which, in this case, gives twenty forward speeds to the spindle. A backing belt is not required, as a thread indicator is attached to the right-hand side of the carriage, as shown in Fig. 1, so that odd, even or fractional threads can be caught with the half nut after returning the carriage by hand when chasing threads. The head is rigid, and is so constructed as to form a reservoir, which permits the gear train to run in oil. The lathe is easily changed from belt to motor drive; a cover with brackets for the mounting of the motor is substituted for the plain cover of the head, and a train of gears or chain drives from the armature shaft to the driving shaft mounted at the back of the lathe.

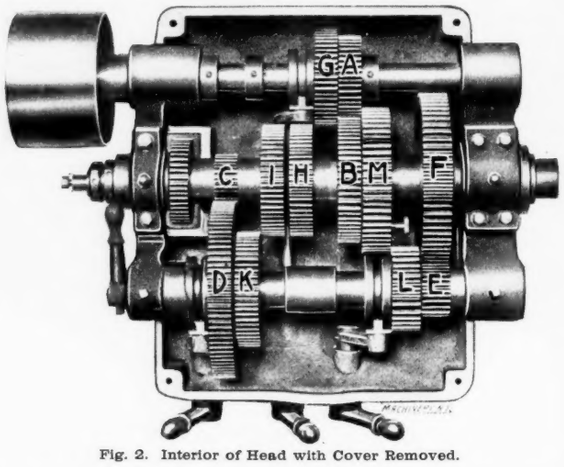


Fig. 2. Interior of Head with Cover Removed.

The tailstock is extra long, is well braced, has set-over screws, a large diameter spindle and is clamped to the bed by four bolts. The forward pair of these bolts are as close to the end of the tail as possible, to counteract the lifting strain of the

cut. The carriage is heavy, has a wide bridge and long bearings, and is usually furnished with a compound rest having a long movement to the top slide. The apron is of box form, and gives all shafts having severe duty bearings on each end. The inner bearing of the rack pinion is bolted to the carriage at the point where it engages in the rack.

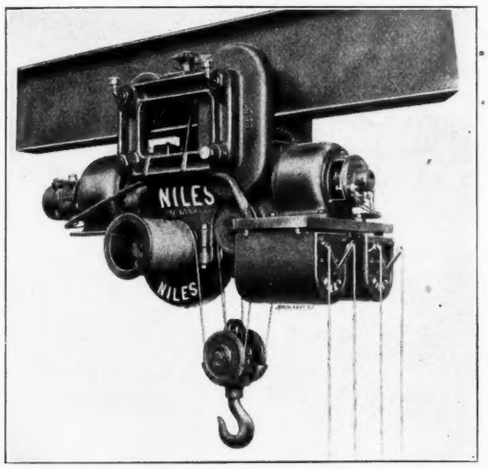
The lathe, as illustrated, gives three changes of feed or screw pitches without change of gears. It can, however, be made with quick change feed device in the apron, giving 40 changes of speed or screw pitches without change of gear. The feed mechanism is so constructed that the operation of any feed locks out all the others. The feed is from a splined leadscrew, the threads of the screw only being used for screw cutting. The cross and the lateral feeds are engaged by the same friction; pulling forward the friction handwheel engages the lateral feed, while the reverse movement engages the cross feed.

This lathe is also made with a double independent tool block with set-over to front tool block for splitting the cut. The patent geared head is furnished in all sizes from 18-inch to 42-inch swing inclusive. These lathes are made by the Draper Machine Tool Co., Worcester, Mass.

NEW NILES ELECTRIC TRAVELING HOIST.

An electric traveling hoist, built by the crane department of the Niles-Bement-Pond Company, is shown in the accompanying illustration. The hoist is compact and self-contained in one heavy cast-iron frame, to which the motors are attached, end on.

The power is transmitted directly from the armature shaft to the drum shaft through one train of worm and worm-wheel



Electric Traveling Hoist.

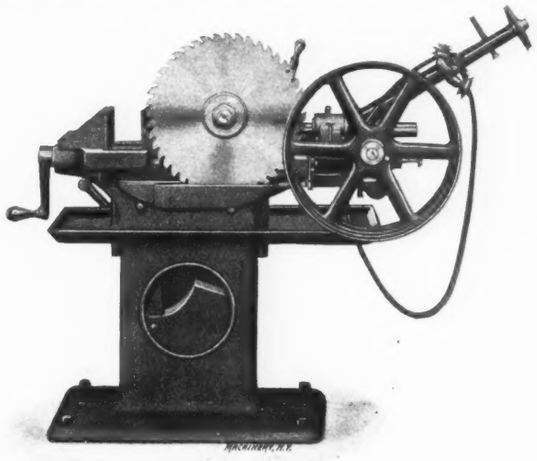
gears. The traversing mechanism is also driven through one train of worm and worm-wheel gears, similar to the hoisting mechanism, except that, when the trolley is arranged to run on a single I-beam, a double set of transmission gears is used. All the mechanism is enclosed in oil and dust-proof casings, and is absolutely noiseless in operation. In addition to the braking effect obtained by the use of the worm and worm wheel, a powerful electric brake is attached to the hoist motor.

These hoists when mounted on a traveling bridge may be used as small capacity cranes. When used as cranes, the hoists are arranged to run between the two I-beams or channels of the bridge, and the controllers for raising and lowering the hook and operating the traversing mechanism may be placed either on the hoist, on the bridge, and operated by cords from the floor, or in an operator's cage attached to the bridge.

The Niles hoists are built in capacities of three-fourths to six tons, and are usually arranged to run on an I-beam track. They will run on straight and curved tracks, and are generally provided with a separate motor for traversing, but if desired, hand traverse may be furnished, or all the traversing mechanism may be omitted and the trolley moved along the track by pushing on the load. The increased service of the electric traverse, however, much more than compensates for the slight additional cost.

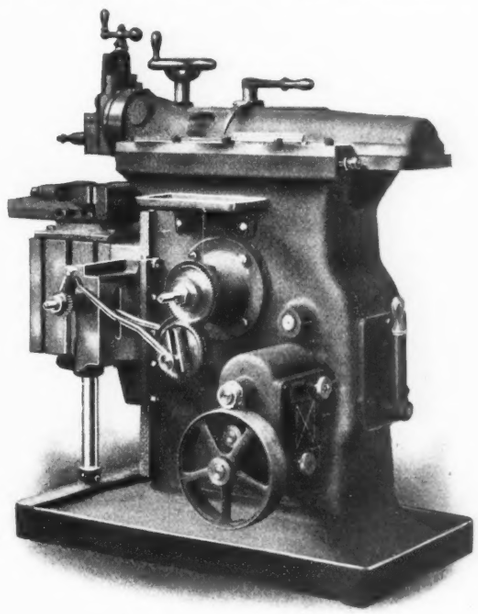
AUTOMATIC COLD SAW CUTTING-OFF MACHINE.

The Hill Standard Mfg. Co., Anderson, Ind., have recently put on the market an automatic cold saw cutting-off machine which we show in the accompanying illustration. With this machine the saw blade is not revolving when work is being put into and removed from the vise; hence there is no danger of the workmen being caught. The revolution of the saw is



Automatic Cutting-off Machine.

started by the advance of the saw blade toward the stock, this action throwing in the clutch. The position of the saw mandrel is so related to that of the axis of the work in the vise that when the stock is almost cut off and begins to weaken the saw cut is opened by gravity, thus preventing any pinching of the blade. The feed is automatically released when the work is cut off, the saw returning to the starting position



Crank Shaper with Gear Box.

and its revolution ceasing at the same time. An automatic saw sharpening attachment by which the saw may be sharpened on the mandrel goes with the machine, thus avoiding the troubles which arise when the saw blade has to be removed to another machine for sharpening and then replaced on the sawing machine. The saw of this cutting-off machine is 14 inches diameter by 5/32 inch thick, and it has a capacity of 3 x 4 inches.

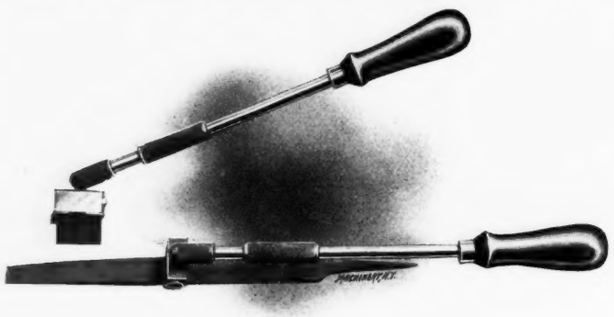
CRANK SHAPER WITH GEAR BOX.

Herewith is illustrated a 16-inch back geared crank shaper with gear box attached. The shaper is one of the standard machines of the Cincinnati Shaper Co., Cincinnati, O., with the cone pulley left off and the gear box attached in its place.

There are three shafts in the box, two of which carry sliding gears; these are moved and locked by the hand nuts shown. This gear box can be applied to any of the company's back geared crank shapers at any time and provides all changes of speed obtainable through the usual cones, and gives further the advantage of a large area of belt contact at all speeds without shifting the belt.

LUTZ UNIVERSAL TOOL HOLDER.

A file holder is shown below by which files and short pieces of files may be conveniently handled for filing grooves, sharp corners and other irregularities where the handling of an ordinary file would be awkward. The holder consists of a vise head containing two movable jaws which hold securely any short piece or whole file. The handle bar is joined to the vise head by a universal ball joint. The vise head can be adjusted

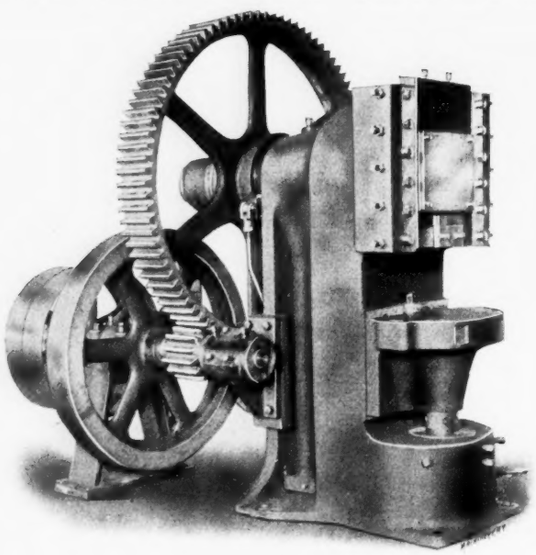


Universal File Holder.

and tightened in any desired position by unscrewing and screwing the handle bar; the latter being provided with a knurled grip. The various working parts of the holder are made of case-hardened steel. The holder may be used for holding regular files for flat and grooved surfaces, round or square short piece files for concave work or square corners, or short piece files either flat or edgewise. The holder is made by the Lutz Tool Mfg. Co., Springfield, O.

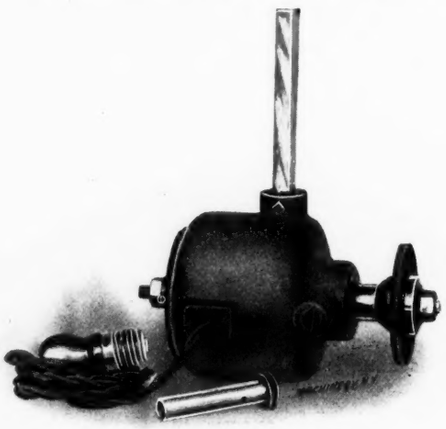
PRESS FOR HEAVY PUNCHING.

A new type of power press adapted for heavy punching, stamping, etc., in the manufacture of fittings for electrical motors and dynamos and other work requiring heavy pressure,



Power Punching Press.

has been built by the E. W. Bliss Co., 5 Adams St., Brooklyn, N. Y., and is shown in the accompanying half-tone. The frame of the press is a solid casting and weighs about 15,000 pounds. The shaft is of large diameter, has a 1¾ inch stroke and is directly connected to the slide without any adjustment, the latter being obtained by raising or lowering the bed, which is supported on a screw of large diameter in addition to being bolted and gibbed to the frame. The screw gives an adjustment of one inch to the table, and is securely locked by a threaded key with two setscrews. The press is controlled by an automatic jaw clutch which positively locks the large gear

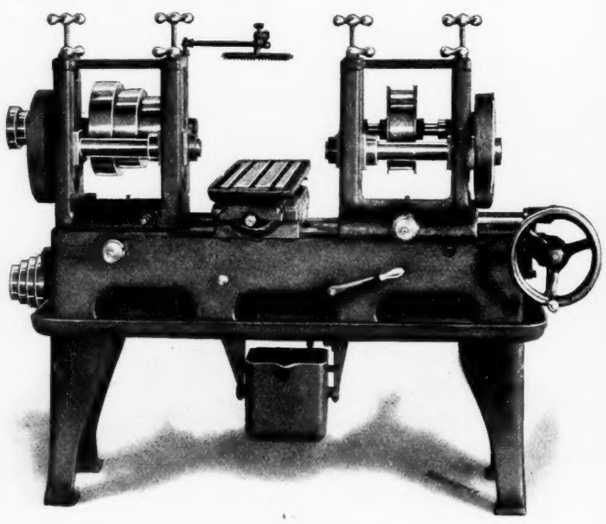


Electrical Lathe Center Grinder.

to the shaft and releases same after the press has made one revolution. There is an extra long slide, and gibs are provided which guide the punches held in them. The bed of the press is 17 x 20 inches, the distance between the gibs is 16 inches and the depth of throat is six inches.

ELECTRICAL LATHE CENTER GRINDER.

Mr. J. F. Willey, Louisville, Ky., is making a new electrical lathe center grinder which is sold by James Clark, Jr., & Co., also of Louisville. In this tool, shown below, the usual slide for feeding the wheel has been eliminated, the feed being attended to by the lathe compound rest upon which the grinder is placed. The compound rest of course can be quickly set to



Double Head Milling Machine.

the desired angle of the center to be ground. When used on lathes equipped only with plain rests, separate cuts are taken equal to the width of the wheel. The bearings of the grinder are conical and adjustable for wear. By leaving out the separate slide feed a more solid construction at less cost is claimed. A high speed is said to be obtained through an improved method of winding.

DOUBLE HEAD LINCOLN MILLING MACHINE.

The Niles-Bement-Pond Co., of New York, have recently brought out a double head milling machine which is shown in the accompanying illustration. This machine is adapted for a similar class of work to that of the single head Lincoln milling machine, but on the double head machine two milling cutters of different diameters may be used, the drive for each

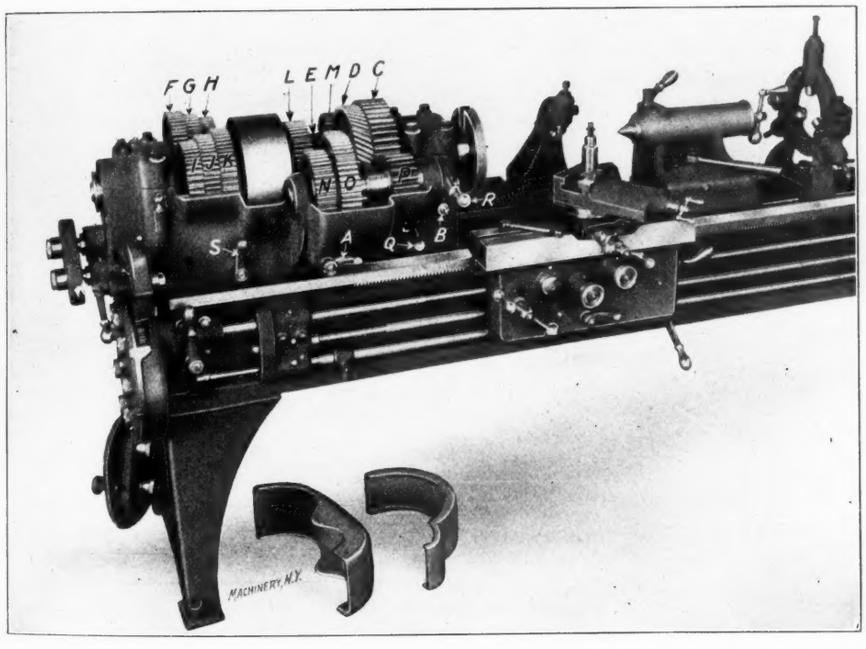
head being independent. On some classes of work this results in a great saving of time over the use of the single head machine. The head shown to the left is fixed and has nine spindle speeds, being driven in connection with a clamp cone pulley of three steps. The movable head to the right is driven through a flanged pulley and has three spindle speeds. For very heavy work both heads may be used to drive the same arbor. The dimensions of the table are 32 inches by 7½ inches, and it has a travel of 15 inches. The table is adjustable laterally 6½ inches, and the spindles may be adjusted vertically 8⅛ inches. Four changes of feed are provided.

RAPID REDUCTION LATHE, NINETEEN-INCH CAPACITY.

Herewith is illustrated a lathe recently designed by the Springfield Machine Tool Co., Springfield, O., having a capacity sufficient to swing a piece 19 inches in diameter in the rough.

The point of greatest interest in this design is the headstock. Twenty spindle speeds are obtainable hereby with a two-speed countershaft; the drive being from a 12-inch single face pulley running direct upon the spindle at moderate speed. A 4½ inch belt is used. These 20 speeds are, first, two direct belt speeds obtainable by a positive clutch operated by handle *A* connecting the driving pulley to the spindle. Secondly, six speeds are obtained when lever *A* is in the position shown and face gear *C* is connected to the spiral gear *D* by a positive clutch operated by lever *B*. The spiral gear *D* is driven by its pinion keyed to the shaft *E* in the rear, this shaft having three gears, *F*, *G* and *H*, mounted loosely upon it and meshing with gears *I*, *J* and *K*, which are keyed fast to the pulley. Any one of the gears *F*, *G* and *H* may be connected to its shaft *E* by an interior positive clutch operated by handle *S*.

Twelve more spindle speeds are obtainable through the double back gears which are placed in front of the spindle. Face gear *C* is disconnected from the spiral gear *D*, allowing the latter to revolve freely upon its spindle. Keyed to spiral gear *D* are two gears, *L* and *M*, either of which may engage back gears *N* and *O*. The drive is then from the pulley through gears *I*, *J* or *K* with *F*, *G* or *H*, through *E* and *D*; and either *L* or *M* with *N* or *O*, and lastly through pinion *P* and face gear *C*. Gears *N* and *O* are positively locked in one or the other driving position by a hook upon lever *Q*, which also affords means for sliding the gears from one position to the other. This lever receives its locking and unlocking motion from handle *R*, which has an eccentric projection extending through the housing in which the back geared shaft is mounted, with an eccentric bush at the other end. By this system eight finishing speeds are provided, namely, two direct belt speeds and six through the spiral gearing, which latter produces a smooth motion. The twelve back geared speeds remaining are for roughing and heavy cuts upon large diameters. The speeds are arranged to give a constant cutting speed for diameters from 1½ to 18 inches. The back gearing is disposed as it is in front of the spindle to relieve the caps of strain.



Springfield Nineteen-inch Lathe.

Nutting Arbor Press.

The carriage has one V and one flat bearing. It has a power cross movement, the cross feed screw having a micrometer collar reading to 1/1,000 of an inch. A clamp is provided for the carriage when facing. The tailstock is heavy, with a large spindle fitted with a number 5 Morse taper center. The spindle is clamped by two bushes, insuring proper alignment of the spindle. The headstock spindle has a 1 9/16 inch hole its entire length and runs in self-oiling bearings. The center corresponds to a No. 5 Morse taper. The apron is double, furnishing two bearings for all studs. The rack pinion has a bearing supporting it close to the rack and may be withdrawn when screw cutting.

The lathe, as illustrated in the cut, is equipped with the rapid change gear mechanism furnished by the makers with their Ideal lathes. This machine may however, be equipped with a positive geared feed, giving six variations of feed, and regular change gears. The countershaft has one tight pulley and two friction pulleys of an improved form.

NUTTING ARBOR PRESS.

The arbor press, shown above, has been recently designed, and is built by Mr. F. J. Nutting, the sales being controlled by the Patterson Tool & Supply Co., Dayton, O. The feature of the open and shut nut, as clearly shown in the half-tone, gives this machine the desirable combination of a screw press having the advantages of a quick moving lever press. The bed of this press is 23 inches long by 12 inches wide, the frame being 27¾ inches high, or mounted on the legs, 4 feet 8 inches. The press will take in work 17 inches in diameter and the travel of the screw is 10½ inches, the distance from the bottom of the screw in the raised position to the notched segment being 11 5/16 inches.

MISCELLANEOUS TOOLS AND APPLIANCES.

The emery wheel dresser made by F. G. Marbach, Medina, O., is of the usual star-wheel type but with the wheels mounted on a slide designed to slip over the rest in front of the emery wheel. The dresser wheel is then fed up against the emery wheel by a cross slide, the main slide being held by one hand while the cross slide is being operated by the other hand of the workman.

An armature disk notching machine for small disks is built by Zeh & Hahnemann, Newark, N. J. The press is designed to do the work ordinarily done by one stroke by combination dies in a larger press, and where the number of disks to be turned out is not large answers every requirement, and the machine itself is much less expensive than the larger presses. There is, besides, a saving in the cost of the dies.

A 16-inch engine lathe has been placed on the market by the Fosdick Machine Tool Co., Cincinnati, O. This lathe is built either with a 5-step or a 3-step cone, the latter if desired for use with high speed steel. It has a modern arrangement of feed gears designed to give 40 changes and a screw cutting range from 2 to 56 threads per inch, and is equipped with compound rest. The main bearing is 2¾ inches in diameter.

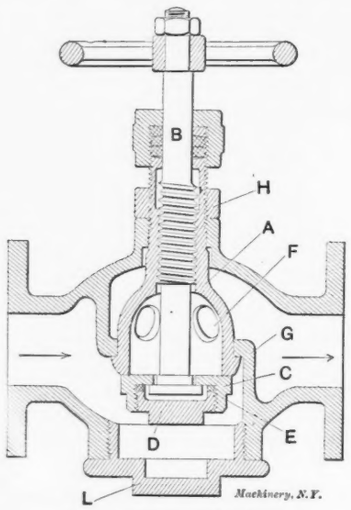
A water tool grinder made by the J. G. Blount Co., Everett, Mass., has a vertical centrifugal pump for flooding the wheel with water during grinding. The pump shaft is driven by means of a friction pulley which bears against the rim of the main driving pulley. The shaft runs in self-oiling bearings placed above the water-line. The pulleys are held in contact by a spring. A large water pan is placed below the wheel and collects the dirt and abrasive material coming from the wheel, and allows it to settle before the water is drained into the tank to which the pump is connected.

A tachometer of novel design is made by the Empire Machine Co., Pittsburg, Pa. The parts are contained in a small air-tight cylindrical case. The mechanism consists of a rotary fan driven by a train of gearing and a vane pivoted on the same axis as the fan, but entirely independent of the fan mechanism. A pointer attached to the vane is normally held at the zero position by a hair spring interposed between the rotary vane and a stationary part of the tachometer. The result is that the speed at which the fan is rotated will determine the position of the fan and the pointer, owing to the air current developed, and thus indicate the speed of the shaft or other rotating element that is being tested.

* * *

VALVE WITH REMOVABLE SEAT.

We show herewith a form of globe valve of novel design, as illustrated in the *Portfeuille Economique des Machines*, for February, 1905. In most valves of this type the valve seat is fixed, so that when the parts become so badly worn that they cannot be reground in place, the whole valve casting must be carried to a machine tool for the necessary machining operations, which necessitates the breaking of the joint between the valve and the pipe.



Valve with Removable Seat.

In the valve shown, both the valve and the seat are removable. The body of the valve is divided into two compartments by a horizontal annular central partition, *G*, the ring being machined to receive the faced ledge or boss of the casting *A*. This piece *A* is screwed into the valve casing, and carries a check nut *H*.

The casting *A*, pierced with openings *F*, is traversed by the valve stem *B*, which carries the valve and is raised and lowered by the hand wheel; the cap placed on the top of *H* assures the tightness of the valve stem in the ordinary manner. The valve, properly speaking, is carried by the stem *B*. It is composed of two parts, *C* and *D*, screwed one into the other, to permit of their mounting upon the stem *B*; an asbestos joint *E* prevents any passage of steam between parts *C* and *D*. It is evident that to remove the pieces liable to wear from the valve, it is only necessary, after raising the hand wheel and removing the cover plate *L*, to unscrew the check nut from the part *A*. These pieces are all made perfectly interchangeable, and the replacing of any worn part should be very easy.

* * *

NEW PRESIDENT FOR LEHIGH UNIVERSITY.

At the meeting of the trustees of Lehigh University held at South Bethlehem, Pa., in June, Henry S. Drinker was elected president to succeed the late Dr. Thomas M. Drown. Mr. Drinker graduated from Lehigh in 1871 as an engineer of mines, and was in 1872 in charge of the construction of the Musconetcong Tunnel. Upon the completion of this, Mr. Drinker spent two years in the preparation of his well-known work upon tunneling, which is still considered the standard in its field. Shortly afterward Mr. Drinker was admitted to the bar. He is at present general solicitor for the Lehigh Valley Railroad, having served in that capacity for many years. Mr. Drinker has long been a trustee and active supporter of Lehigh University. His selection for the presidency, the acceptance of which entails personal sacrifices for Mr. Drinker, is eminently satisfactory to the trustees and alumni of the university.

* * *

CONCERNING NAME PLATES.*

So far as its operation is concerned, a dynamo or motor would be just as well off without a name plate as with one; yet there is probably no other part of the machine that is examined so carefully by visitors to a station or other interested persons not employed therein. And it is a matter of reasonable pride to the electrical engineer, being an indication of the thoroughness and precision with which his work has been done, that every piece of revolving and transforming apparatus for which he is responsible carries upon it this certificate of output ability, voltage, current, speed, etc. Of course, this is not the reason that name plates are put on dynamos, motors, converters, transformers, etc.; the chief object is to inform everyone interested in the matter just what the apparatus is intended to do, what its maximum output ability is and who made it (possibly the last-mentioned point should have been put first in the list). This method of imparting the information mentioned is not employed solely for the benefit of casual visitors, either; it is of prime importance to the operating staff of a station. It is not rational to expect employes to memorize the characteristics of every machine or important piece of apparatus about the plant, although most of them in time do so involuntarily, and it is highly essential that the fundamental data relating to any piece of apparatus shall be immediately available when required; the name plate meets this requirement perfectly—when it is properly made.

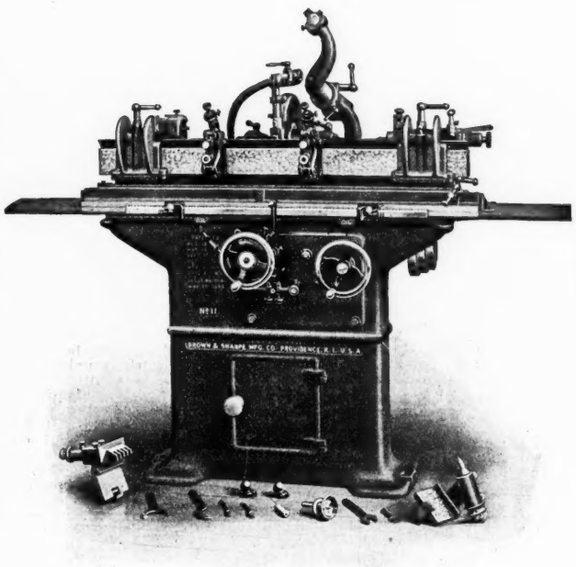
The foregoing argument applies just as forcibly to steam apparatus in an electric lighting or power plant as to the electrical equipment, yet no one ever saw a boiler name plate on which was stamped the number, diameter and length of the tubes, size of the grate, total area of heating surface, evaporating ability, etc.; nor does an engine name plate give the sizes

* *American Electrician*, June, 1905.

Brown & Sharpe Mfg. Co.

PROVIDENCE, R. I., U. S. A.

Economy in Time, Accuracy of Product



2 Features

that adapt the

No. 11 Plain Grinding Machine

To meet the Requirements of Manufacturers.

OTHER FEATURES:

No water guards required.

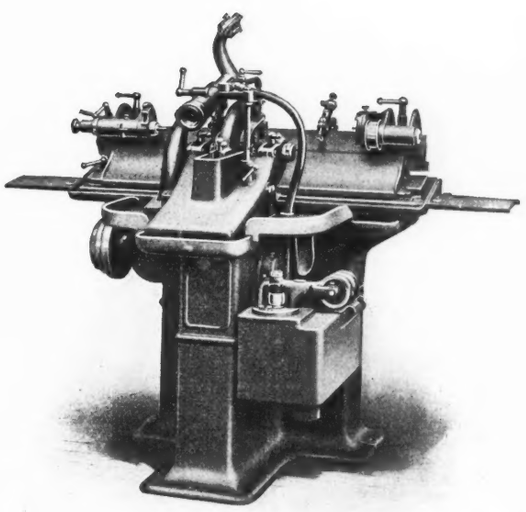
Automatic cross feed sizes accurately to one quarter of a thousandth.

Provision for abundant supply of water.

Universal back rests for long, slender work.

Adjustments easily and quickly made.

Accuracy of feeding mechanism allows work to be ground close to shoulder.



Send us samples or specifications of work and we will furnish estimates.

CATALOG AND DESCRIPTIVE CIRCULARS FREE.

of the cylinders, most economical point of cut-off, rating in pounds of steam per horse-power-hour at this cut-off and a stated initial pressure, or any other technical information beyond the so-called horse-power and preferred speed. In the case of pumps, the makers have gone a little further ahead as to name plates and usually put on the cylinder sizes; it would be of considerable assistance to the user if the capacity in gallons per minute at the preferred speed were also put on, and the pressure for which the pump is designed might advantageously be included. The name plates on pumps for fire service are models of satisfaction in these particulars.

The point is that a great deal of valuable time is often lost by having to hunt up or calculate data which ought to be plainly stamped on the name plate of a piece of machinery; moreover, in the handling of machines equipped with incomplete name plates mistakes are made which would easily be avoided otherwise and which in some cases cause the machine to suffer disastrously for the petty sin of its builder.

* * *

FRESH FROM THE PRESS.

THE POCKETBOOK OF REFRIGERATION AND ICE MAKING. Edited by A. J. Wallis-Tayler. Published in America by the Norman W. Henley Publishing Co., 132 Nassau Street, New York. 184 pages, illustrated, pocketbook size. Price, $1.50.

The contents of this pocketbook comprise, among other matter, an outline of the subject of refrigeration; cold storage; ice making and the storing of ice; insulation; the testing and management of refrigerating machinery; general tables and memoranda. It was prepared with the idea of providing a pocket book which should contain such formulas, tables and data, with brief descriptive matter, as would prove useful to persons interested in refrigeration and cold storage. The matter is, to a large extent, in tabular form.

FERRIC AND HELIOGRAPHIC PROCESSES: A HANDBOOK FOR PHOTOGRAPHERS AND DRAFTSMEN. By George E. Brown. Published in America by Tennant & Ward, 287 Fourth Avenue, New York. 149 12mo. pages. Price, $1.00.

This little handbook is designed to serve two classes of people. First, amateur photographers, with a taste for experiment, who may like to make their own sensitive papers, and draftsmen, who have to make prints from tracings. It will doubtless appeal to the first class more in England, where the book is published, than in this country, but the information upon making various kinds of prints from tracings supplies a want in draftsmen's literature that, so far as we know, has not heretofore been met. The book first describes blueprinting processes and methods for toning blueprints to produce other colors than blue. Then follow the printing of blue lines on white ground, and the printing of black and brown lines on white ground, or white lines on black or brown ground. There are 22 chapters in all, going into the subject in much detail.

ALTERNATING CURRENT MACHINERY. By William Esty, S. B. and M. A., head of Department of Electrical Engineering, Lehigh University: American School of Correspondence, Armour Institute of Technology. Chicago: 1905. Pages, 412. Price, $3.75.

This volume has been prepared with the especial object of giving practical electricians and beginners a working knowledge of alternating current apparatus without the necessity of the readers understanding the higher mathematics. A rudimentary knowledge of algebra and trigonometry should be sufficient for the clear comprehension of the principles treated. The descriptions, explanations and proofs are simple and clear; graphical or geometric methods rather than analytical or algebraic have been adopted wherever possible. Many examples and numerical illustrations have been used throughout the text to reenforce the principles by their practical application. The reader is supposed to have some acquaintance with the simpler laws of electricity and magnetism.

The student is first acquainted with the essential features of the source of alternating currents, the alternator, and with the features of alternating current problems which differentiate them from direct current problems. After this a description of alternating current measuring apparatus follows. Several chapters are devoted to synchronous motors, the transformer, the rotary converter, the induction motor, station appliances and lightning arresters. In these descriptions the theory, applications, operations and structural detail of each class are described and illustrated.

The volume is well bound, printed on good paper, and is fully illustrated.

MACHINE TOOLS AND WORK SHOP PRACTICE. By Alfred Parr, Instructor in Fitting and Machine Shop, and Forge, University College, Nottingham; with an introduction by Wm. Robinson, Professor of Engineering, University College, Nottingham. 432 pages 5½ x 8½ inches. Illustrated with 462 cuts. Published by Longmans Green & Co., London and New York. Price, $4.00.

The introductory remarks by Prof. Robinson are of interest, especially that in which he speaks of the accuracy of modern measuring instruments. The assertion is made that with the ordinary Whitworth measuring machine differences of 1-100,000 inch or about 1-4,000 millimeter can be detected, but that the electric micrometer enables us to measure small dimensions and magnitudes of the order of 1-5,000,000 millimeter or 1-120,000,000 inch. This instrument reveals elevations on a plane surface or errors in a screw thread which no machine yet made can remove; an attempt at removal with ordinary means produces a hollow. The book takes up progressively the following subjects, one chapter for each: Measurement; Marking or Lining-Out and Processes; Materials; Drills and Drilling Machinery; Turret Lathes; Treadle and Power Lathes; Lathe Appliances; Boring Machines; Lathe Work; Grinding Wheels and Machines; Cold Iron and Steel Sawing Machines—Planing, Shaping and Slotting Machines; Milling Machines, Milling Cutters and Appliances; Gearing and Gear Cutters; Cutting Tools and How to Use Them; Fitting, Erecting and Vise Work and Tools; Forge Shop Tools and Forging Processes; Hydraulic Machine Tools and Riveting; Transmission of Power; Methods of Working; Appendix.

It should be obvious from the nationality of the author that the work treats generally of English practice, although this is by no means universally the case as a number of American machine tools are described and American small tools such as milling cutters and hand tools such as micrometers, levels, depth, gages, etc., are shown of American pattern. In fact in all works on English practice that we have examined the illustrations of small tools are almost invariably of the American pattern, and one is puzzled to know what small tools are representative of English manufacture, if any are. The chapter on measurement is of much interest, although we can hardly credit the statement made in the introductory notes by Prof. Robinson on the marvelous refinement now possible by electric touch. In this chapter a very good resume is given of the Brown & Sharpe Mfg. Co.'s duplication of the standard yard by Mr. Oscar J. Beale, and the chapter, as a whole, is a c. apliment to the work of Brown & Sharpe in the matter of fine measuring instruments. Lack of space prohibits a thorough review of the work and we must limit ourselves to the general statement that it is of general worth to almost anyone interested in machine shop practice and the general art of manufacturing.

NEW TRADE LITERATURE.

In the last number of MACHINERY the price list of "Boldi Steel" sold by Peter Frasse & Co., New York, was referred to; it should have been "Poldi Steel."

THE WASHBURN SHOPS, Worcester, Mass. Circular of adjustable drawing stands for draftsmen, students, etc.

THE BULLARD MACHINE TOOL CO., Bridgeport, Conn. Circular entitled "One Day's Work," showing what can be done in turning piston rings on a Bullard mill.

HYATT ROLLER BEARING CO., Harrison, N. J. Bulletin containing fac-simile letters from prominent users, expressing approval of the Hyatt roller bearings and giving reasons therefor.

MARSHALL & HUSCHART MACHINERY CO., 62-64 So. Canal St., Chicago, Ill. Illustrated sheets containing descriptions of various machine tools handled by this firm; the sheets are perforated and designed to be bound into covers for their preservation and convenient reference.

THE ELECTRIC CONTROLLER & SUPPLY CO., Cleveland, O. Catalogue of lifting magnets. These magnets are of various designs according to the kind of material and the weight of material handled, and the catalogue contains information upon these several types and explains the practical points connected with their use.

C. W. HUNT CO., 45 Broadway, New York City. Catalogue No. 054 upon manila rope transmission and hoisting. This pamphlet, besides being a catalogue of the rope transmission systems built by this company, is also a treatise on the subject of rope transmission, containing valuable data by Mr. C. W. Hunt.

WELLMAN-SEAVER-MORGAN CO., Cleveland, O. Illustrated catalogue of coal-handling machinery. This is one of the finest catalogues we have received and it explains in detail and with numerous fine engravings the remarkable progress that has been made in recent years in handling coal at the mines, docks and in storage.

YALE & TOWNE MFG. CO., 9-15 Murray St., New York. Two catalogues, one dealing with the Yale & Towne chain blocks and electric hoists and the other with locks and hardware for railroad use, especially for railroad cars and stations. A feature of the first pamphlet is a folder map of New York City, showing the route of the subway.

THE MORTON MFG. CO., Muskegon Heights, Mich. Catalogue of draw-cut shapers. These well-known shapers have been developed along original lines for numerous classes of work, ranging from ordinary work that is done on pillar shapers to heavy railroad work and floor-plate work requiring portable tools. This catalogue describes these various lines and illustrates many of the machines.

BROWN & SHARPE MFG. CO., Providence, R. I. Illustrated circular of their new No. 12 Automatic Gear Cutting Machine, which is illustrated in this number of MACHINERY. Also, illustrated circular of the new machinists' tools, including a micrometer set in sizes up to 6 inches, steel beam trammels, universal dividers, straightedges, and scribers.

CINCINNATI MACHINE TOOL CO., Cincinnati, O. Catalogue of upright drilling and tapping machines. Drills manufactured by this company are made in sizes from 21 to 42 inches and in a variety of styles. They are built either with or without reversing arrangement for the spindle to be used in tapping. Certain styles are fitted with an improved positive geared feed with quick change.

CROCKER-WHEELER CO., Ampere, N. J. Illustrated circular of railway generators, comprising a description of various installations and data upon generators in sizes up to 1,500 K. W. Also, illustrated pamphlet with a complete description of the electrical equipment of the Pittsburg & Lake Erie R. R., written by Roy V. Wright, formerly mechanical engineer of these shops and now associate editor of the *American Engineer and Railroad Journal.*

INTERNATIONAL STEAM PUMP CO., 114 Liberty St., New York. A large pamphlet describing a complete line of steam, electric, gas and power-driven compressors of different types is being distributed by the Laidlaw-Dunn-Gordon branch of this company. One of the features of this pamphlet is a notice of the improved system of air-valve gears. It also explains the air-lift for raising water from driven or bored wells and gives much engineering data.

MORSE TWIST DRILL & MACHINE CO., New Bedford, Mass. Pamphlet entitled "Young Machinist's Practical Guide." This contains 45 pages devoted to tables of practical use to machinists, especially in regard to drills and milling cutters. There are numerous other tables, however, such as the weight of metals, data upon threads, etc. This will be sent to any one sending his name and address, but it is requested that the full address be clearly written since their experience has been, in their previous offers to machinists, that the addresses are frequently insufficient to enable them to fill the order.

NORTHERN ELECTRICAL MANUFACTURING CO., Madison, Wis., have issued Bulletin No. 44, which illustrates representative applications of their back-geared motors in industrial plant service. Slow speed for all sorts of machines is readily provided by Northern back-geared motors. The back-geared attachment is a compact, self-contained device which turns the driving shaft at slow speed without the necessity of employing a motor designed for slow armature speed. In other words, the ordinary high-speed armature is employed, which means a motor of low cost for slow speed machines.

CINCINNATI MILLING MACHINE CO., Cincinnati, O. Illustrated "Treatise on Tool Room Grinding and Grinding Machines," containing a great deal of valuable information upon the subject of cutter and tool grinding. Part 1 is devoted to the No. 1 Universal Cutter and Tool Grinder which has been manufactured for a long time; and Part 2 to the No. 2 Grinder, which is a new one and is a complete universal grinder for cutter and tool grinding from the smallest cutter up to face mill 24 inches diameter. Numerous illustrations show how to handle a great variety of work and the description in connection with the illustrations gives valuable information upon cutter grinding, applicable to all such work regardless of the make of machine used. The book is sent free of charge to those interested in the subject.

The Lackawanna has just issued, through its Industrial Department, a little booklet containing comprehensive information about factory buildings either new or vacant or available for manufacturing and industrial enterprises, located in territory served by that road in New York, New Jersey and Pennsylvania. A description is given of each building, its present equipment, and whether or not it is served by private sidetrack, together with some general information as to the town in which it is located. For those contemplating the organization of a new industry or the re-location of one already established, the book contains many suggestions that may prove of interest. Copies of this booklet may be had free upon application to Wendell P. Colton, Industrial Agent, 26 Exchange Place, New York City.